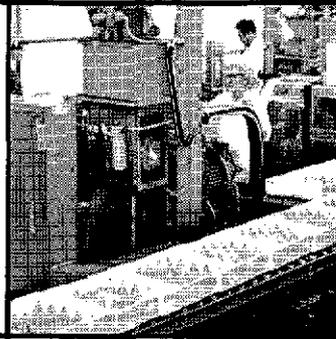
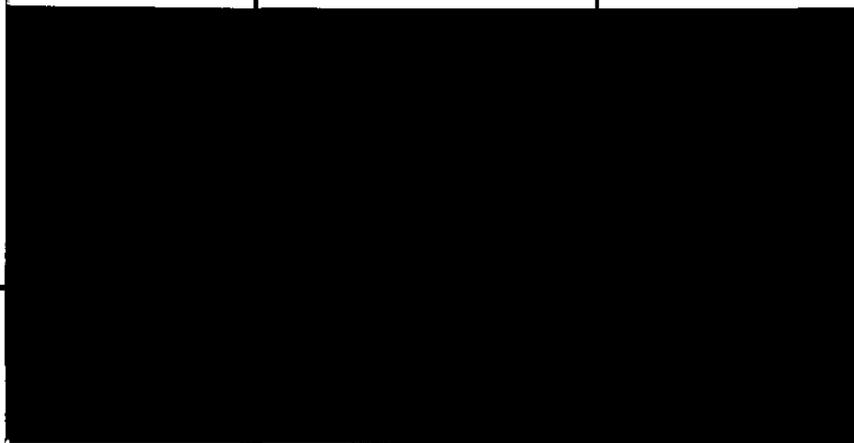
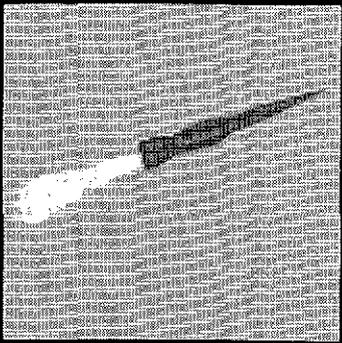


NASA CR-

141614



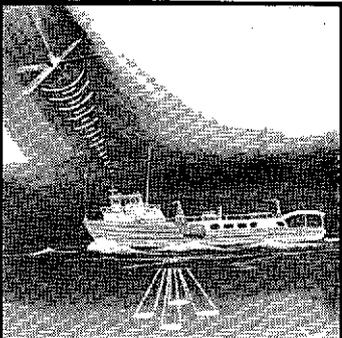
(NASA-CR-141614) OMS ENGINE SHUTOFF VALVE
AND ACTUATION SYSTEM DESIGN AND EVALUATION
Final Report (Marquardt Corp.) 202 p HC
\$7.25

N75-16621

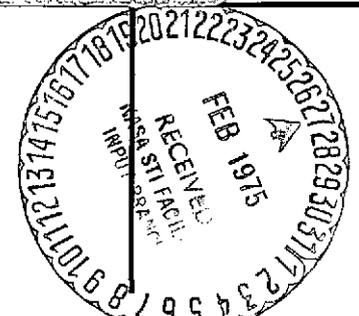
CSCL 21H

Unclas

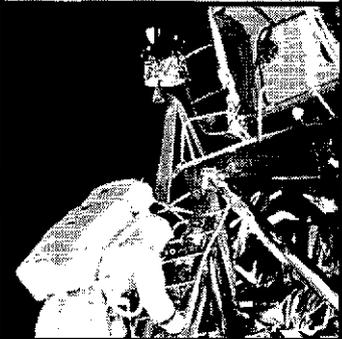
G3/20 - 09692



THE
Marquardt
COMPANY



VAN NUYS, CALIFORNIA



Contract NAS9-13443
DRLT 891 Item 5
DRD MA-129T

S1316

**OMS ENGINE SHUTOFF VALVE
AND
ACTUATION SYSTEM
DESIGN AND EVALUATION**

FINAL REPORT

December 15, 1974

Prepared for:

**National Aeronautics and Space Administration
Lyndon B. Johnson Space Center
Houston, Texas 77058**

H. Wichmann

**The Marquardt Company
16555 Saticoy Street
Van Nuys, California 91409**

ABSTRACT

A tradeoff study was performed to determine those shutoff valve and actuation system concepts that are most suitable for the OMS engine application. During this tradeoff, emphasis was placed on the ten year and 100 mission life requirement, propellant and propellant residue compatibility and weight. These requirements made it apparent that poppet or ball valves utilizing electric or electropneumatic actuation were most applicable. Preliminary design layouts of a number of valve and actuation concepts were prepared and analyzed to make the optimum concept selection. A significant ground rule in this comparison was the fact that pneumatic actuation systems were required to feature their own pneumatic supply. Thus for the quad redundant valve, it was necessary to include two pneumatic supply systems, one for each of the series legs of the quad redundant package. The requirement for the pneumatic package placed heavy reliability, weight, and maintenance penalties upon electropneumatic actuation systems. Consequently, the two most promising valve and actuation systems concepts selected featured electric torque motor operation and a poppet as well as a ball valve concept with a retractable seal.

Both the electric torque motor operated dual poppet valve and the electric torque motor operated ball valve featuring a solenoid retracted seal were detail designed during this program. In addition, a flow fixture simulating the dual poppet valve flow path was also detail designed. The flow fixture and the electrically operated ball valve were subsequently fabricated and tested.

Pressure drop tests of the dual poppet valve flow fixture substantiated predicted pressure drop characteristics (0.6 psi at 5 pounds per second water). The electrically actuated ball valve was life cycled 10,000 cycles utilizing a breadboard electronic driver for the brushless DC torque motor. The valve successfully completed this test program without any significant performance degradation. Thus, the design verification test program demonstrated the superior performance capability of the electrically actuated ball valve concept in comparison with electro-pneumatic concepts with respect to weight, reliability, and maintainability throughout the space shuttle program.

TABLE OF CONTENTS

<u>Section</u>	<u>Title</u>	<u>Page</u>
	Abstract	ii
	Foreword	iii
1.0	Summary	1-1
2.0	Introduction	2-1
3.0	Technical Requirements	3-1
3.1	General	3-1
3.1.1	Study Requirements	3-1
3.1.2	Design Requirements	3-1
3.1.3	Development Requirements	3-1
3.1.4	Technical Guidelines	3-1
3.2	Technical Changes Implemented During the Performance of the Program	3-4
3.2.1	Scope Changes	3-4
3.2.2	Pressure Drop	3-4
4.0	Analysis and Preliminary Design	4-1
4.1	Preliminary Valve Tradeoff	4-1
4.1.1	Background	4-1
4.1.2	OME Quadredundant Valve Configuration	4-5
4.1.3	Pneumatic Supply Package	4-5
4.1.4	Valve Descriptions	4-8
4.1.5	Conclusions	4-18
4.2	Final Valve Tradeoff	4-20
4.2.1	Description of Final Tradeoff Candidates	4-20
4.2.2	Valve Analysis Approach	4-26
4.2.3	Final Tradeoff Conclusions	4-51
4.3	Valve Pressure Drop Tradeoff	4-53
5.0	Detail Design	5-1
5.1	Dual Poppet Valve	5-1
5.2	Ball Valve	5-15
5.2.1	Valve Operational Description	5-16
5.2.2	Construction Details and Materials of Construction	5-19
5.2.3	Performance Characteristics	5-21
5.3	Dual Poppet Valve Flow Fixture	5-29
6.0	Fabrication	6-1
6.1	Fabrication of the Dual Poppet Valve Flow Fixture	6-1
6.2	Bipropellant Ball Valve and Actuation System Fabrication	6-3

TABLE OF CONTENTS (continued)

<u>Section</u>	<u>Title</u>	<u>Page</u>
7.0	Testing	7-1
7.1	Dual Poppet Valve Flow Fixture Testing	7-1
7.2	Bipropellant Ball Valve and Actuation System Tests	7-1
7.2.1	Torque Motor/Driver Testing	7-1
7.2.2	Bipropellant Valve and Actuation System Tests	7-11
8.0	Recommendations and Conclusions	8-1
9.0	References	9-1
Appendix A	Force Balance OMS Engine Shutoff Valve Double Poppet Type	A-i
Appendix B	Ball Valve Stress Analysis	B-i
Appendix C	Parts List Prototype Ball Valve	C-i
Appendix D	Test Plan MTP 0200	D-i

LIST OF ILLUSTRATIONS

<u>Figure</u>	<u>Title</u>	<u>Page</u>
4-1	Quadredundant Valve Block Diagram	4-6
4-2	Pneumatic Supply System Schematic	4-7
4-3	In-Line Pneumatic Operated Poppet Valve	4-10
4-4	Pneumatically Operated Poppet Valve	4-11
4-5	Pneumatically Operated Ball Valve	4-13
4-6	Torque Motor Operated Poppet Valve	4-15
4-7	Torque Motor Operated Ball Valve	4-16
4-8	Dual Poppet Valve	4-21
4-9	Ball Valve - External Gear Seal Lift-Off	4-23
4-10	Ball Valve - Internally Actuated Seal Lift-Off	4-24
4-11	Seal Lift-Off Mechanism	4-25
4-12	Ball Valve With Solenoid Retracted Seal	4-27
4-13	Valve Analysis Approach	4-28
4-14	Filter ΔP Vs. Effective Filter Screen Area	4-29
4-15	Effective Flow Area Vs. Percent Valve Open	4-31
4-16	Typical Torque Motor Phase Control Logic	4-39
4-17	Torque Motor Digital Driving Signals	4-40
4-18	Torque/Speed at 40 Watts	4-41
4-19	DC Torque Motor Characteristics - Stall Torque Vs. No Load Speed	4-42
4-20	DC Torque Motor Characteristics - Motor Weight Vs. Stall Torque	4-42
4-21	Balanced Poppet Valve Maximum Force Required Vs. Stroke	4-43
4-22	Weight of Gear Box Vs. Gear Ratio	4-44
4-23	Motor Weight Vs. Gear Ratio	4-44
4-24	Actuation System Weight Vs. Gear Ratio	4-45
4-25	Valve Opening Response Vs. Torque Motor Input Power	4-46
4-26	Ball Valve Maximum Force Required Vs. Stroke	4-47
4-27	Ball Valve Torque Motor Stall Torque Required Vs. Gear Ratio	4-48
4-28	Weight of Ball Valve Gear Box Vs. Gear Ratio	4-48
4-29	Ball Valve Torque Motor Weight Vs. Gear Ratio	4-49
4-30	Ball Valve Weight of Actuation System Vs. Gear Ratio	4-49
4-31	Valve Opening Response Vs. Torque Motor Input Power	4-50
4-32	Series Redundant Valve Design Layout	4-56
4-33	Quadredundant Valve Design Layout	4-57
4-34	Quad Weight Vs. Quad Valve Package ΔP	4-58
5-1	OME Poppet Valve	5-2
5-2	Dual Poppet Valve Elevation View	5-3
5-3	Poppet Valve Actuation Sequence	5-5
5-4	Lead-Lag Capability Baseline Requirement - 100 Ms Fuel Lead Poppet Valve Opening Motion	5-6

LIST OF ILLUSTRATIONS (Continued)

<u>Figure</u>	<u>Title</u>	<u>Page</u>
5-5	Poppet Valve Actuation Forces Bipropellant Valve	5-8
5-6	Torque Motor Characteristics, Part Number X29580	5-10
5-7	Poppet Valve Opening Response	5-11
5-8	OME Ball Valve	5-14
5-9	Ball Valve Cross Sections	5-17
5-10	Ball Valve Actuation Sequence	5-18
5-11	Ball Valve Actuation Torques Bipropellant Valve	5-22
5-12	Lead-Lag Capability Ball Valve Opening Motion	5-23
5-13	Ball Valve Opening Response	5-25
5-14	Dual Poppet Flow Test Fixture	5-29
6-1	Dual Poppet Valve Flow Fixture	6-2
6-2	Breadboard Electronic Driver	6-4
6-3	Ball Valve Assy. Prototype	6-5
6-4	OME Ball Valve	6-7
6-5	Bipropellant Ball Valve Assembly	6-9
6-6	Actuator Gear Box	6-10
6-7	Actuator - Exploded View	6-12
6-8	Valve Components	6-13
6-9	Solenoid Assembly	6-14
7-1	Poppet Valve Flow Fixture Test Set-Up	7-2
7-2	Dual Poppet Valve Pressure Drop Vs. Water Flow Rate	7-3
7-3	Dual Poppet Valve Pressure Drop Vs. Water Flow Rate And Open Position	7-4
7-4	Hall Effect Device Output Vs. Time	7-6
7-5	Hall Effect Device Output Voltage and Rectified Driving Current Vs. Time (Phase II)	7-6
7-6	Driving Voltage Vs. Time	7-7
7-7	Torque Motor No Load Speed Vs. Voltage	7-7
7-8	OME Valve Test Setup Schematic	7-8
7-9	OME Valve Test System	7-9
7-10	OME Valve Inlet and Outlet Plumbing	7-10
7-11	Pressure Drop Vs. Flow Rate Oxidizer Valve Only	7-12
7-12	Response Vs. Voltage (Dry Operation)	7-12
7-13	Response Vs. Pressure Water Flowing, 35 VDC Runs 6-17	7-13
7-14	Torque Vs. Motor Turns, Motor Deenergized - Solenoids Energized	7-13
7-15	Response Vs. Voltage (Wet Operation)	7-15
7-16	Oscillograph Trace - Life Cycle Program (2 Cycles Shown)	7-17
7-17	Response Vs. Accumulated Cycles	7-18
7-18	Leakage Vs. Accumulated Cycles Oxidizer Valve	7-18
7-19	Torque Motor Temperature Rise Curve 70°F Ambient Temperature	7-20

LIST OF TABLES

<u>Table</u>	<u>Title</u>	<u>Page</u>
4-I	Past Engine Shut Off Valve and Actuation System Problems	4-2
4-II	Valve Element - Motion Characterization	4-4
4-III	Valve and Actuation System Trade Off	4-4
4-IV	Pneumatic Supply Tank Sizing Criteria	4-8
4-V	Pneumatic Supply System Weight Breakdown	4-9
4-VI	Pneumatic Supply System Maintenance Requirements	4-9
4-VII	Preliminary Valve Trade Off Summary	4-19
4-VIII	Candidate Materials for Space Shuttle OME Shutoff Valve	4-34
4-IX	Dual Poppet Valve and Actuation System Weight Breakdown	4-46
4-X	Ball Valve Weight and Actuation System Breakdown	4-51
4-XI	Final Trade Off Data Comparison	4-52
4-XII	OME Propellant Shutoff Valve Final Tradeoff	4-54
4-XIII	Dual Poppet Valve Comparison Quad Valve Package Vs. Series Valve Package	4-58
5-I	Brushless DC Torque Motor Specifications	5-8
5-II	Poppet Valve Weight	5-13
5-III	Performance Characteristics Quadredundant Dual Poppet Valve	5-15
5-IV	Solenoid Design Criteria	5-26
5-V	Ball Valve Weight	5-27
5-VI	Performance Characteristics - Quad Redundant Ball Valve P/N L14091	5-28

FOREWORD

This report is submitted by The Marquardt Company in accordance with the requirements of NASA Contract NAS 9-13443. The work was administered by the NASA Lyndon B. Johnson Space Center, Houston, Texas, with Mr. J. Fries as the NASA Technical Project Manager.

This program was performed by the Engineering Department of The Marquardt Company at the Van Nuys facility. The Program Manager was Mr. H. Wichmann and Project Engineer was Mr. D. Slagle. Other contributors to this program were Messrs. T. L. Kelly, A. Malek, I. Dickens, A. Maderian, E. Austin, O. F. Anderson, R. Dickinson, L. Beman, and R. F. Williams.

1.0 SUMMARY

The objective of NASA Contract NAS 9-13443 was the development of shutoff valve and actuation system technology for the engine of the orbital maneuvering system of the Space Shuttle. This valve and actuation system requires longer service life, greater contamination tolerance, greater propellant compatibility, and improved maintainability than currently available valve and actuation systems. The program included an assessment of the current state-of-the-art of valves and actuation systems and the determination of the deficiencies of the existing designs, particularly those developed for the Apollo program. Based on this assessment and understanding, more advanced valve and actuation systems capable of meeting the Space Shuttle requirements were defined. Both a dual poppet valve and a ball valve featuring a retractable seal were determined to be the optimum valve concepts in combination with an actuation system consisting of a brushless DC torque motor and driver. Bipropellant valves, including actuation systems, of both concepts were detail designed. In addition, a flow fixture simulating the flow passages through the dual poppet valve was also detail designed. This flow fixture and the bipropellant ball valve and actuation system were subsequently fabricated and subjected to a test evaluation program. The test program substantiated the soundness of the design approach and verified its suitability for the Space Shuttle OMS engine application.

The initial task of the program served to gain an understanding of the problems encountered with previously developed valve and actuation systems and to evaluate these designs with respect to the Space Shuttle requirements of multimission, long service life. Particular emphasis was placed on understanding the problems encountered with the valve and actuation systems during the Apollo program. The primary deficiencies of these valves and actuation systems were determined to be a lack of contamination tolerance, propellant compatibility, maintainability, and service life. In particular, the ball valves used on the Apollo engines featured sliding seals which degraded rapidly as a function of cycle life and which were easily damaged by the presence of particles or residues on the ball surface. The actuation systems used in conjunction with these valves were of the electro-pneumatic type which, as far as the Space Shuttle requirement was concerned, meant the inclusion of a complete pneumatic supply system with its inherent reliability, maintainability, and weight penalties. Several materials used in these valves were not considered suitable for a long-term propellant exposure and particularly for potential accommodations of the propellants and moisture. A most apparent example of this lack of compatibility was the use of aluminum on the oxidizer side.

During the performance of the first task of this program, a number of preliminary valve and actuation system designs which minimized or eliminated potential problems for the Space Shuttle application were prepared. The performance characteristics of these concepts were determined and a comparison was made to arrive at the optimum valve and

1.0 SUMMARY

The objective of NASA Contract NAS 9-13443 was the development of shutoff valve and actuation system technology for the engine of the orbital maneuvering system of the Space Shuttle. This valve and actuation system requires longer service life, greater contamination tolerance, greater propellant compatibility, and improved maintainability than currently available valve and actuation systems. The program included an assessment of the current state-of-the-art of valves and actuation systems and the determination of the deficiencies of the existing designs, particularly those developed for the Apollo program. Based on this assessment and understanding, more advanced valve and actuation systems capable of meeting the Space Shuttle requirements were defined. Both a dual poppet valve and a ball valve featuring a retractable seal were determined to be the optimum valve concepts in combination with an actuation system consisting of a brushless DC torque motor and driver. Bipropellant valves, including actuation systems, of both concepts were detail designed. In addition, a flow fixture simulating the flow passages through the dual poppet valve was also detail designed. This flow fixture and the bipropellant ball valve and actuation system were subsequently fabricated and subjected to a test evaluation program. The test program substantiated the soundness of the design approach and verified its suitability for the Space Shuttle OMS engine application.

The initial task of the program served to gain an understanding of the problems encountered with previously developed valve and actuation systems and to evaluate these designs with respect to the Space Shuttle requirements of multimission, long service life. Particular emphasis was placed on understanding the problems encountered with the valve and actuation systems during the Apollo program. The primary deficiencies of these valves and actuation systems were determined to be a lack of contamination tolerance, propellant compatibility, maintainability, and service life. In particular, the ball valves used on the Apollo engines featured sliding seals which degraded rapidly as a function of cycle life and which were easily damaged by the presence of particles or residues on the ball surface. The actuation systems used in conjunction with these valves were of the electro-pneumatic type which, as far as the Space Shuttle requirement was concerned, meant the inclusion of a complete pneumatic supply system with its inherent reliability, maintainability, and weight penalties. Several materials used in these valves were not considered suitable for a long-term propellant exposure and particularly for potential accommodations of the propellants and moisture. A most apparent example of this lack of compatibility was the use of aluminum on the oxidizer side.

During the performance of the first task of this program, a number of preliminary valve and actuation system designs which minimized or eliminated potential problems for the Space Shuttle application were prepared. The performance characteristics of these concepts were determined and a comparison was made to arrive at the optimum valve and

actuation system configurations. The optimum concepts were determined to be a dual poppet valve and a ball valve featuring a solenoid retracted seal, both with a brushless DC torque motor actuation system. The performance characteristics of these two concepts were determined for both the nominal pressure drop characteristics, as well as, one-fourth of the nominal pressure drop characteristics.

Detail design drawings of prototype versions of both of the selected concepts were subsequently prepared. The dual poppet valve featured a contamination tolerant teflon/metal sealing closure interface, redundant teflon jacketed dynamic shaft seals, and full metallic, flexure guidance. Thus, the valve mechanism was completely friction free and promised high cycle life and low leakage characteristics. The ball valve also featured a teflon/metal sealing closure interface. The ball valve seal was retracted by means of a solenoid prior to ball movement such that no scrubbing whatsoever occurred at the sealing closure interface. In addition, the ball shaft was located eccentrically to the sealing closure interface such that the clearance between the ball and the sealing closure continually increased as the ball was being opened. The ball was located by means of teflon impregnated bearings. Redundant teflon jacketed seals were employed as the dynamic seals at the ball shafts. Both the ball and poppet valves included purge passages which utilized the dynamic pressure of the purge gases to route them to relatively dead volumes and to thereby thoroughly purge all portions of the valve.

Both the dual poppet valve and the ball valve were designed to operate with brushless, direct current torque motors. This type of actuation system offers very low weight, high reliability, simplicity, and ease of operation (zero maintenance). For the operation of the brushless DC torque motor, a breadboard electronic driver was designed. The drive train between the torque motor consisted of three gear elements in the case of the ball valve and two gear elements and a cam in the case of the poppet valve. Both gear trains were designed to feature a fuel lead during valve opening and fuel lag during valve closing. Except for the torque motors themselves, the actuators including the gear trains, were designed to be made of teflon impregnated aluminum; thus, all bearing surfaces in the actuator were teflon lubricated and required no other type of lubrication. The actuating systems also included position indicators for the monitoring of the valve position. Detailed design drawings of the flow fixture simulating the flow passage through the dual poppet valve were also prepared.

Both the dual poppet valve flow fixture and the bipropellant ball valve and actuation system were subsequently fabricated. Upon completion of the fabrication effort, the pressure drop characteristics of the dual poppet valve flow fixture were determined to be as originally predicted (0.6 psi at 5.95 lbs/second of nitrogen tetroxide). The bipropellant ball valve and actuation system was subjected initially to integration tests of the breadboard electronic integration driver and the torque motor and subsequently

to performance testing and life cycle testing of the complete valve and actuation system on a water flow bench. Over 10,000 life cycles were accumulated on the valve and actuation system without any malfunctions or any significant performance degradation. Opening valve response characteristics of 500 to 800 milliseconds, depending on operation pressure, were demonstrated, although closing responses were slower than required. Ball valve pressure drop characteristics were 0.2 psi at 5.95 lbs/seconds of nitrogen tetroxide. Leakage tests performed throughout the life cycle program which included both dry and wet cycling were determined to be negligible for the shaft seals and to be generally less than one SCIM through the oxidizer valve. Test data and post test inspection revealed several minor design problems, but these are readily correctable. In light of the substantial number of advanced design features, the test program was considered very successful.

The OMS engine shutoff valve and actuation system design and evaluation program resulted in the demonstration of a valve concept which is 35% lighter than the present baseline electropneumatic concept of the Space Shuttle program. In addition, the tested torque motor-operated bipropellant ball valve concept promises greatly improved contamination resistance, maintainability, reliability, and service life. Implementation of this valve and actuation system approach into the Space Shuttle program will substantially reduce Space Shuttle costs and improve Space Shuttle reliability.

THIS PAGE LEFT INTENTIONALLY BLANK

2.0 INTRODUCTION

The OMS engine valve and actuation system technology program described in this report was performed in support of the Space Shuttle Program of the National Aeronautics and Space Administration. The Space Shuttle vehicle is designed to provide low cost transportation to earth orbit to support a variety of missions, including logistics resupply of a space station. To achieve maximum cost effectiveness, the Space Shuttle is being designed for up to 100 flights (reuses) over a 10-year operational life time, including the capability to relaunch within two weeks after earth landing. The system is being designed to minimize required post flight refurbishment, maintenance and checkout, and for simplicity and ease of maintenance when required. For translational maneuvers, the Space Shuttle employs two rocket propulsion systems called the orbital maneuvering systems (OMS). These propulsion systems are pressure fed rocket systems employing nitrogen tetroxide and monomethylhydrazine as the propellants and helium as the pressurant. The valve and actuation system technology developed during this program was specifically intended for the shutoff valve located on the OMS engine and constituting an integral part of the OMS engine assembly.

A need for the performance of the valve and actuation system technology program described herein resulted from deficiencies inherent in other spacecraft engine valves, particularly those employed during the Apollo program, to satisfactorily perform during the long-life, multimission requirements of the Space Shuttle. These deficiencies generally consisted of the lack of sufficient propellant compatibility and contamination tolerance, valve reliability for extended cycle-life, and zero maintenance characteristics. Therefore, the valve and actuation system design philosophy and accepted design practices of past applications had to be extrapolated and new approaches developed for the OMS engine valve and actuation system of the Space Shuttle to eliminate these deficiencies.

The OMS engine shutoff valve and actuation system design and evaluation program consisted of six principle tasks. These are:

1. Analysis and preliminary design
2. Detailed design
3. Prototype valve and actuation system fabrication
4. Prototype valve and actuation system testing
5. Flow fixture evaluation
6. Final report

The purpose of task No. 1 was the determination of the optimum valve and actuation system concept for the Space Shuttle OMS engine shutoff valve application. A survey was conducted to define the failure modes, development problem areas, and operational problems in previous valve and actuation system development programs, with emphasis on programs that imposed requirements for earth storable propellant compatibility, extended service life, low contamination sensitivity, and other critical areas pertinent to the OMS shutoff valve program. Based on Marquardt's experience, a compilation of theoretical or potential problem areas was prepared and was combined with the survey data to establish a comprehensive definition of the potential problem areas to be encountered in the development of the OME propellant shutoff valve and actuation system. Design approaches were developed to resolve or minimize each of the problem areas. Based upon the contractual, technical guidelines, and the survey of problem areas, the Marquardt Company prepared preliminary valve and actuation system designs for both quadredundant and series redundant valves with emphasis on the bipropellant valves making up each of these multiple valve configurations. The inherent design, performance, and operating characteristics of each of these concepts were determined. Comparisons of the development costs, unit cost, weight, and envelope, as well as inherent life, maintainability, and contamination sensitivity were established between the concepts. A final comparison of the performance characteristics of the selected concepts based on the nominal pressure drop requirements and one-fourth of the nominal pressure drop requirements were also made.

During task No. 2, the detail design of the two selected valve and actuation systems was performed. This design effort included detail drawings to the piece part level, detail drawings depicting the maintainability of the design, and overall assembly drawings. In addition, a parts and material list was prepared. Differences between the prototype hardware and the flightweight design were carefully evaluated to ensure that the prototype valve and actuation system performance was representative of the flightweight valve design. To minimize program design and fabrication costs, the detail design drawings were concerned only with a bipropellant valve and actuation system, rather than the quadredundant assembly. In addition, design drawings of various test and manufacturing fixtures were also completed.

Task No. 3 served to fabricate and procure all parts required for the assembly of one bipropellant valve and actuation system.

During task No. 4, the electrical torque motor-operated bipropellant ball valve was subjected to a test evaluation program. This program was defined in a detail test plan and consisted of the evaluation of the torque motor as a component, as well as, performance and life-cycle tests of the valve and actuation system on a water flow bench. All data obtained during this program were reduced, compiled, and analyzed for subsequent inclusion in the final report.

Task No. 5 was performed to substantiate that the pressure drop through the dual poppet valve, which was the second valve and actuation system concept designed during task two, could be met. Consequently, a flow fixture simulating the flow path through one dual poppet/seat interface was designed, fabricated, and tested. Pressure drop characteristics, as a function of poppet stroke were measured.

Task No. 6 served to identify the effort in writing this final report.

The OMS engine shutoff valve and actuation system design and evaluation program resulted in the feasibility demonstration of an electric motor-operated bipropellant ball valve whose features are such as to eliminate problem areas encountered with previous designs. This program resulted in the demonstration of major performance improvements in the areas of weight, reliability, maintainability, compatibility with propellant residues, and life. Implementation of the demonstrated valve and actuation system design will result in a major cost savings to the Space Shuttle Program.

THIS PAGE LEFT INTENTIONALLY BLANK

3.0 TECHNICAL REQUIREMENTS (as copied from the contract)

3.1 GENERAL

3.1.1 Study Requirements

The contractor shall conduct trade studies and conceptual design efforts to identify advanced valve and actuation system design approaches suitable to meet the long life, maintainability, and economic development requirements for the OMS engine bipropellant shutoff valve. One valve and actuation system concept will be selected for experimental evaluation and design verification. The selected valve and actuation system concept shall be suitable for packaging and use in a mechanically linked quadredundant shutoff valve configuration and also in a valve configuration, which incorporates only series redundancy. A detailed design with appropriate analysis and drawings will be established for a lightweight shutoff valve. A prototype test valve design shall also be established which contains all the essential design features to experimentally verify fabrication and operation of the lightweight shutoff valve configuration. The prototype test valve need not be to the same level of redundancy as the lightweight valve design, if analysis indicates that all essential valve and actuation system features can be experimentally verified with a simpler test configuration.

3.1.2 Design Requirements

The contractor will define in detail the concepts and theories emanating from the study effort. Environmental conditions under which the valves and actuation systems will satisfactorily operate and the performance and detailed characteristics of the equipment will be clearly specified.

3.1.3 Development Requirements

The contractor will specify those special factors that must be considered in translating the design data into tangible end items. The contractor should identify any problems which become evident and might potentially affect manufacturing processes and techniques. The solutions to these problems should identify what must be developed in order to facilitate manufacturing of the end product. The contractor will conduct testing and prepare test documentation to verify that the performance design requirements of the valve(s) and actuation system(s) meet the requirements of this SOW.

3.1.4 Technical Guidelines

The following guidelines, with a few noted exceptions, are not to be considered firm requirements. They are intended as optimum design objectives and are subject to change in accordance with technology limitations and reliability considerations. One of the primary objectives of this contractual effort is to define the realistic and obtainable requirements that should be imposed on a valve and actuation system for the Space Shuttle OME and thus hopefully avoid development problems that may result from initially unrealistic performance requirements.

3.1.4.1 Application

The valve and actuation system technology and design recommendations developed as a result of this contractual effort will be utilized in defining the recommended design, operational capabilities, and requirements for the OME propellant valve and actuation system.

3.1.4.2 Fluid Media Compatibility

The valve for this program must be compatible for exposure to the following propellant vapors, liquids, and combinations of oxidizer and fuel vapors. The propellants will be nitrogen tetroxide (N_2O_4), 50/50 blend of hydrazine and unsymmetrical dimethylhydrazine (50% N_2H_4 - 50% UDMH), and monomethylhydrazine (MMH). The contractor will have conclusive compatibility data on each material recommended for usage. In evaluating propellant compatibility, the contractor will also evaluate propellant moisture combinations since once a valve is exposed to propellants it is unreasonable to assume that the unit will remain free of moisture for the remaining service life. The contractor will not consider propellant decontamination of components to extend the service life, since cleaning of hardware between missions is improbable and will result only when required to insure personnel safety during system repairs. The valve must also be compatible with anticipated cleaning and flushing fluids.

3.1.4.3 Lubricants

Due to propellant compatibility, low temperature operation, and extended service life, total exclusion of lubricants is a desirable design goal.

3.1.4.4 Maintainability

The valve must be designed to be easily maintained. Replaced detail part of the valve must not affect the operational characteristics of the valve.

3.1.4.5 Checkout

The valve and actuation system should be designed to minimize actuation of the valve for checkout purposes.

3.1.4.6 Cycle Life

A design goal of 4000 wet cycles and 6000 dry cycles will be used for this program.

3.1.4.7 Internal Leakage

A leak rate of 10 standard cubic centimeters per hour (scch) of helium will be used as a goal.

3.1.4.8 Pressure Drop

A maximum pressure drop of 5 psid from the valve inlet to the valve outlet which will include all filters and redundant valves. The valve design shall provide for a balanced pressure drop in the event of a failure in one of the parallel flow paths to minimize the resulting engine mixture ratio shift.

3.1.4.9 Response

The valve opening and closing times shall have absolute actuation times in the range of 100 to 1000 milliseconds. The actual times will be established through trade studies considering valve actuation approaches and the effects of valve actuation times on the engine start and shutdown transient.

3.1.4.10 Response Repeatability

Response repeatability should be considered an important factor in the design of the valve and actuation system.

3.1.4.11 Filters

Filters used in the valve and actuation system should be consistent with the contamination tolerance of the valve and actuation system.

3.1.4.12 Fabrication Limitations

In the process of designing a prototype valve to satisfy the requirements of this SOW, the contractor should maintain an awareness of the design requirements that will be imposed on a "flight-type" design to insure that the prototype will be adaptable.

3.1.4.13 Weight and Envelope

Minimum weight and envelope are important design considerations not to be overlooked by the contractor.

3.1.4.14 Contamination

Contamination tolerance will be a major design objective for this program. Limitation of self-generated contamination shall also be a primary design goal.

3.1.4.15 Decontamination

Dead-ended passages, crevasses, and other possible areas which contaminations could collect and hinder a decontamination process should be avoided.

3.2 TECHNICAL CHANGES IMPLEMENTED DURING THE PERFORMANCE OF THE PROGRAM

3.2.1 Scope Changes

Upon completion of the trade-off studies, the program was redirected to permit detail design of two valve and actuation systems and design, development, and test of a flow fixture.

3.2.2 Pressure Drop

The total quad valve pressure drop requirement was reduced from 5 psid to 1.25 psid at nominal flow rate upon completion of the trade-off studies.

4.0 ANALYSIS AND PRELIMINARY DESIGN

The objective of this task was to generate a number of preliminary valve and actuation system designs which were capable of meeting the technical requirements of the OME valve and actuation system as described in section 3.0 and to compare these designs and select the most promising one. This objective was accomplished by initially performing a preliminary valve tradeoff involving six valve and actuation system configurations. From these six configurations the two most promising configurations were selected and a final valve tradeoff was performed. This final tradeoff included a design study of three different means of retracting the seal in a ball valve and the detailed sizing and analysis of the two selected configurations. Upon completion of the final valve tradeoff one more analytical investigation was performed to determine the response, power, weight, and envelope degradation that would be incurred if the pressure drop requirement was reduced by a factor of 4. The resulting predicted degradation was determined to be acceptable and the subsequent detail design effort of the two selected valve concepts was therefore performed on valves featuring the lower pressure drop.

4.1 PRELIMINARY VALVE TRADEOFF

4.1.1 BACKGROUND

The preliminary tradeoff basically consisted of two parallel efforts. One effort was a survey to define the failure modes, development problem areas, and operational problems in previous valve and actuation system programs and the other effort consisted of the listing and review of possible valve configurations. These two efforts were then combined to permit the selection of the most applicable valve and actuation system design concepts and to permit a final tradeoff between these concepts.

To make certain that the valve concepts considered during the tradeoff study would be fully compatible with the space shuttle requirements and would not include inherent weak features employed in existent propellant shutoff valves, a review of the existing shutoff valve and actuation systems was made. This included such valves as the transtage, LM Ascent, LM Descend, and SPS Engine valves. In reviewing these designs particular emphasis was placed on their suitability to meet the performance requirements listed in section 3 such as earth storable propellant compatibility, extended service life, low contamination sensitivity, minimum maintenance, etc. A summary of the problems identified during this review is presented in Table 4-I. This table is divided into three columns which list the problem, cause, and the corrective design approach that was employed during the preparation of preliminary designs of the valve concepts used in the tradeoff study.

The principal problems identified in Table 4-1 are sealing closure leakage and corrosion. Sealing closure leakage was generally due to the presence of particles or due to sealing closure interface surface finish degradation resulting from the scrubbing that occurred during opening and closing. The presence of particles at the sealing enclosure resulted from a number of causes. These included self generated contamination due to the rubbing of adjacent internal valve parts, foreign particles resulting from a lack of sufficient cleaning of the valve or the system or a lack sufficient filtration in the system, and particles resulting from propellant residues and chemical reactions (corrosion). This corrosion was generally

TABLE 4-1

PAST ENGINE SHUT OFF VALVE AND ACTUATION SYSTEM PROBLEMS

PROBLEM	CAUSE	CORRECTIVE DESIGN APPROACH
<ul style="list-style-type: none"> ● SEALING CLOSURE LEAKAGE 	<ul style="list-style-type: none"> ● SEALING CLOSURE WEAR DUE TO SCRUBBING ● SEALING CLOSURE WEAR DUE TO PRESENCE OF PARTICLES ● SURFACE FINISH DEGRADATION DUE TO MATERIAL INCOMPATIBILITY WITH OPERATING FLUIDS OR GALVANIC ACTION ● LACK OF PROPER SEALING CLOSURE DESIGN ● CONTAMINATION 	<ul style="list-style-type: none"> ● DESIGN FOR MINIMUM SCRUBBING DURING MATING ● MINIMIZE IMPACT FORCES DURING MATING ● ELIMINATE SEAL DRAG BY LIFTING OFF SEAL ● MINIMIZE SELF GENERATED CONTAMINATION BY ELIMINATING SLIDING FITS AND EMPLOYING HARD MATERIALS IN IMPACT AREA ● PROVIDE INLET SCREEN FOR LARGE (150μ) PARTICLES ● ACHIEVE CLEAN LIFT OFF OF SEAT WITHOUT ANY DRAG ● SELECT COMPATIBLE MATERIALS ● SELECT MATERIALS WHICH ELIMINATE GALVANIC COUPLES ● USE PROVEN SEALING CLOSURE CONCEPTS ● EMPLOY CONTAMINATION RESISTANT SEALING CLOSURE DESIGNS
<ul style="list-style-type: none"> ● FAILURE TO OPEN OR CLOSE AND LACK OF RESPONSE REPEATABILITY 	<ul style="list-style-type: none"> ● INSUFFICIENT FORCE MARGIN IN ACTUATOR ● ICING IN PNEUMATIC VENT LINES ● "STICKTION" IN MOVING PARTS DUE TO PARTICLE CONTAMINATION IN SMALL CLEARANCES ● MECHANISM WEAROUT 	<ul style="list-style-type: none"> ● PROVIDE 100% FORCE MARGIN ● USE NON-WETTING SURFACES IN VENT SECTION ● PROVIDE CHECK VALVES IN VENT SECTION ● REPLACE PNEUMATIC WITH ELECTROMECHANICAL ACTUATION ● ELIMINATE SMALL CLEARANCES THROUGH USE OF FLEXURES ● INCREASE CLEARANCE THROUGH USE OF FLUTES ● EMPLOY DESIGNS THAT DO NOT REQUIRE TIGHT CLEARANCES ● MINIMIZE SELF-GENERATED CONTAMINATION ● ELIMINATE WEARING SURFACE BY MEANS OF FLEXURES OR ENHANCE LIFE BY EMPLOYING LOW FRICTION SLIDING INTERFACE
<ul style="list-style-type: none"> ● CORROSION 	<ul style="list-style-type: none"> ● LACK OF PROPER IDENTIFICATION OF OPERATING FLUIDS AND THEIR COMBINATION WITH MOISTURE AND FLUSHING FLUIDS AS WELL AS WITH ONE ANOTHER ● INSUFFICIENT COMPATIBILITY DATA ● INADEQUATE PURGING ● CREVICE CORROSION 	<ul style="list-style-type: none"> ● CAREFULLY CONSIDER ALL OPERATING ENVIRONMENTS AND OPERATING MODES ● USE ONLY MATERIALS HAVING KNOWN COMPATIBILITY ● PREPARE EASILY PURGEABLE DESIGNS ● ELIMINATE CREVICES
<ul style="list-style-type: none"> ● EXTERNAL LEAKAGE 	<ul style="list-style-type: none"> ● VIBRATION ● TEMPERATURE EFFECTS (STEADY STATE AND TRANSIENT) ● PRESSURE SURGES (UPSTREAM OR DOWNSTREAM) 	<ul style="list-style-type: none"> ● DESIGN FOR ABSOLUTE MINIMUM MOVEMENT AT SEALS ● ANALYZE THERMAL EXPANSION CHARACTERISTICS ● CORRELATE MATERIAL PROPERTIES (PARTICULARLY OF PLASTIC AND ELASTOMERS) AT HIGH AND LOW TEMPERATURES ● USE REDUNDANT SEALS ● USE POSITIVE SEALS SUCH AS BELLOWS OR WELD JOINTS ● PROVIDE ADEQUATE STRESS MARGINS FOR PEAK PRESSURES ● CONTROL RESPONSE TO MINIMIZE SURGES
<ul style="list-style-type: none"> ● POSITION INDICATOR FAILURE 	<ul style="list-style-type: none"> ● VIBRATION ● LACK OF ADEQUATE DESIGN OR SWITCH SENSITIVITY 	<ul style="list-style-type: none"> ● EMPLOY HIGHLY VIBRATION RESISTANT INDICATORS SUCH AS ANALOG DEVICES AND REED SWITCHES ● INSTALL INDICATOR IN LOCATION WHERE LARGE MOVEMENT CAN BE MONITORED AND SWITCH SENSITIVITY IS NOT CRITICAL

4-2

the result of a lack of proper identification of the operating fluids and their combination with moisture and flushing fluids as well as with one another. Since the engine valves are open at the downstream side to the thrust chamber assembly which in turn contains either propellant or both propellants and their reaction products as well as the ambient, moist salt air a wide variety of conditions can exist and these were frequently not taken into consideration sufficiently. None of the valves reviewed were subject to the requirement of maintenance free ten year and one hundred mission service life and consequently no problems relating to this requirement were listed in the failure reports. Nevertheless the designs were reviewed to assess their capability of meeting the extended service life requirement and any negative design features inherent in the existing shutoff valves were avoided during the preparation of the preliminary valve design concepts for this study.

The Marquardt Company has performed a number of valve technology programs for various government agencies. These programs have resulted in the systematic evaluation of potential rotary and linear shutoff devices and actuators as shown in Table 4-II (References 1 and 2). The applicability of various valve elements was again evaluated with respect to the OME valve and actuation system requirements. As discussed in Section 3 these requirements include a number of qualitative requirements such as lubrication, maintainability, contamination, and decontamination which effect the type of valve concept that can be selected. Thus the requirement for no lubrication was desirable since the space shuttle ten year life requirement involving both in space and atmospheric environments has not previously been demonstrated with any lubrication system. Consequently such techniques as teflon impregnated bearing surfaces and friction free metallic flexure guidance were considered design techniques which would eliminate the need for lubrication and these design features in turn were available in certain valve elements but not in others.

In the area of compatibility it was specified that the valve materials should not only be compatible with the propellants themselves but also with flushing fluids which might be employed during decontamination procedures as well as with moisture in combination with the propellants which might result during condensation. In particular the concern with the possible reaction of propellant residues with the moist salt air did not exist during the Apollo program because of the single mission requirement; however for the space shuttle program this is indeed a very real requirement. The possible formation of salts on the downstream sides of both the fuel and oxidizer valves due to the reaction of the propellants with the moist air during the space shuttle program was considered to be highly probable and therefore valve sealing closure designs were generated which are relatively insensitive to the presence of salts. In particular it was recognized that it was necessary to employ lift off type seals rather than the sliding valve sealing closures utilized in the designs of existing propellant valves. The requirement for decontamination capability caused designers at the Marquardt Company to concentrate on valve elements which either eliminated propellant traps or which provided the necessary purge passages to assure that the propellants would be removed from areas that otherwise would have constituted propellant traps.

The application of the OME valve and actuation system requirements to the valve elements listed in Table 4-II resulted in the conclusion that the preliminary valve tradeoff should concentrate on certain sealing closures and actuators which held the greatest promise of meeting the performance requirements. Table 4-III summarizes the particular sealing

TABLE 4-II

VALVE ELEMENT - MOTION CHARACTERIZATION

MOTION TYPE	LINEAR	ROTARY
SHUT OFF DEVICE	POPPET PLUG GATE SLIDING SPOOL BLADE DIAPHRAGM OR BOOT	BALL BUTTERFLY SWING FLAPPER
ACTUATOR	CYLINDER SOLENOID PIEZOELECTRIC MAGNETOSTRICTIVE THERMAL EXPANSION LINEAR MOTOR	TORQUE MOTOR ELECTRICAL MOTOR HYDRAULIC OR PNEUMATIC MOTOR INERTIA WHEEL

TABLE 4-III

VALVE AND ACTUATION SYSTEM TRADE OFF

SEALING CLOSURES	ACTUATORS
POPPET/SEAT <ul style="list-style-type: none"> ● FLAT INTERFACE ● WITH & WITHOUT PRESSURE BALANCING 	PNEUMATIC PISTON <ul style="list-style-type: none"> ● RACK & PINION LINKAGE ● LEVER LINKAGE ● DIRECT DRIVE
BALL <ul style="list-style-type: none"> ● WITH MECHANICAL SEAL LIFT OFF ● WITH SOLENOID SEAL LIFT OFF 	ELECTRIC TORQUE MOTOR <ul style="list-style-type: none"> ● BRUSH TYPE OR BRUSHLESS ● WITH BALL SCREW DRIVE ● WITH GEAR DRIVES ● WITH COMBINATION GEAR & LEVER DRIVES
	SOLENOID ACTUATOR <ul style="list-style-type: none"> ● MAGNETIC LINKAGE ● MECHANICAL LINKAGE

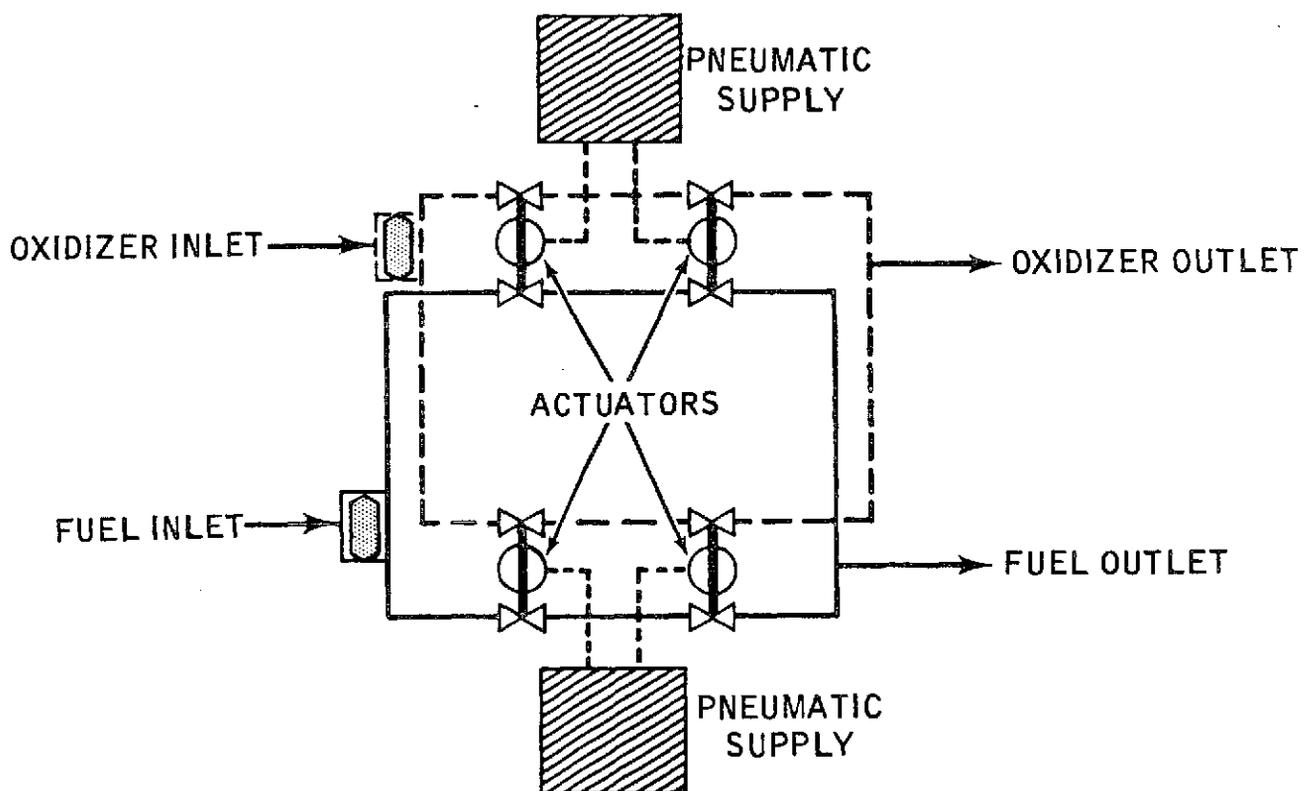
closures and actuators as well as the required actuator linkages that were considered during the preliminary tradeoff study. The ball and poppet type seal enclosures were specified because they appeared most suitable for a lift off sealing closure design which was considered necessary for this application and because they inherently feature low leakage characteristics as required here. Potential means for actuating these sealing closures included pneumatic piston, electric torque motor, and solenoid actuator drives. Propellant operation actuators were not included because of the hazards involved in dumping the propellant from the actuation system.

4.1.2 OME QUAD REDUNDANT VALVE CONFIGURATION

The quadredundant OME shutoff valve and actuation system that was considered during the tradeoff studies is shown schematically in Figure 4-1. The quadredundant valve consists of four bi-propellant valves arranged in series/parallel configuration. A single bi-propellant valve consists of an oxidizer valve which is mechanically linked to a fuel valve and which utilize a single common actuator. Both electromechanical and electropneumatic actuation was considered. A significant ground rule during the evaluation of various valve concepts was the requirement that if pneumatic power was utilized, a separate pneumatic supply for the two series bipropellant valves was required as shown in Figure 4-1. Therefore two pneumatic supply packages were required for the quadredundant valve assembly. As shown in the block diagram a single fuel inlet and a single oxidizer inlet each featuring an inlet screen considered. During subsequent tradeoff and detail design activities it became apparent however that it would be easier for the engine developer to determine the inlet and outlet propellant feed system configuration to the bipropellant valves. Consequently inlet screens were incorporated into each upstream bipropellant valve inlet and the quadredundant valve package in effect featured two oxidizer inlets, two fuel inlets, and similarly two oxidizer outlets and two fuel outlets. Furthermore during OMS engine tradeoff studies it became apparent that it would be more convenient to utilize a counterflow arrangement in the valve such that the fuel and oxidizer through any one bi-propellant valve were flowing in opposite directions. This counterflow arrangement was therefore subsequently used during this program during the detail design effort of the bi-propellant valves.

4.1.3 PNEUMATIC SUPPLY PACKAGE

The requirement for a pneumatic supply package for the two bi-propellant valves in each of the quadredundant valve parallel legs made it necessary to prepare a pneumatic supply package configuration and to determine its performance characteristics. Figure 4-2 shows the pneumatic supply system schematically as it was used during the preliminary tradeoff study. This system is very similar in configuration to that utilized for the SPS engine valves during the Apollo program. The purpose of the pneumatic supply system is to store sufficient gaseous nitrogen to permit the actuation of the two pneumatic actuators of the two bi-propellant valves during the thirty day mission. The system stores gaseous nitrogen at a pressure of 2000 psi and utilizes a pressure regulator to reduce the GN₂ storage pressure to 250 psi for pneumatic actuator operation. Three way solenoid valves control the pneumatic pressure to each of the 2 pneumatic actuators. For reliability reasons a solenoid isolation valve is also included in the pneumatic system in the event one of the three way solenoid valves should develop excessive leakage during



QUADREDUNDANT VALVE BLOCK DIAGRAM

4-7

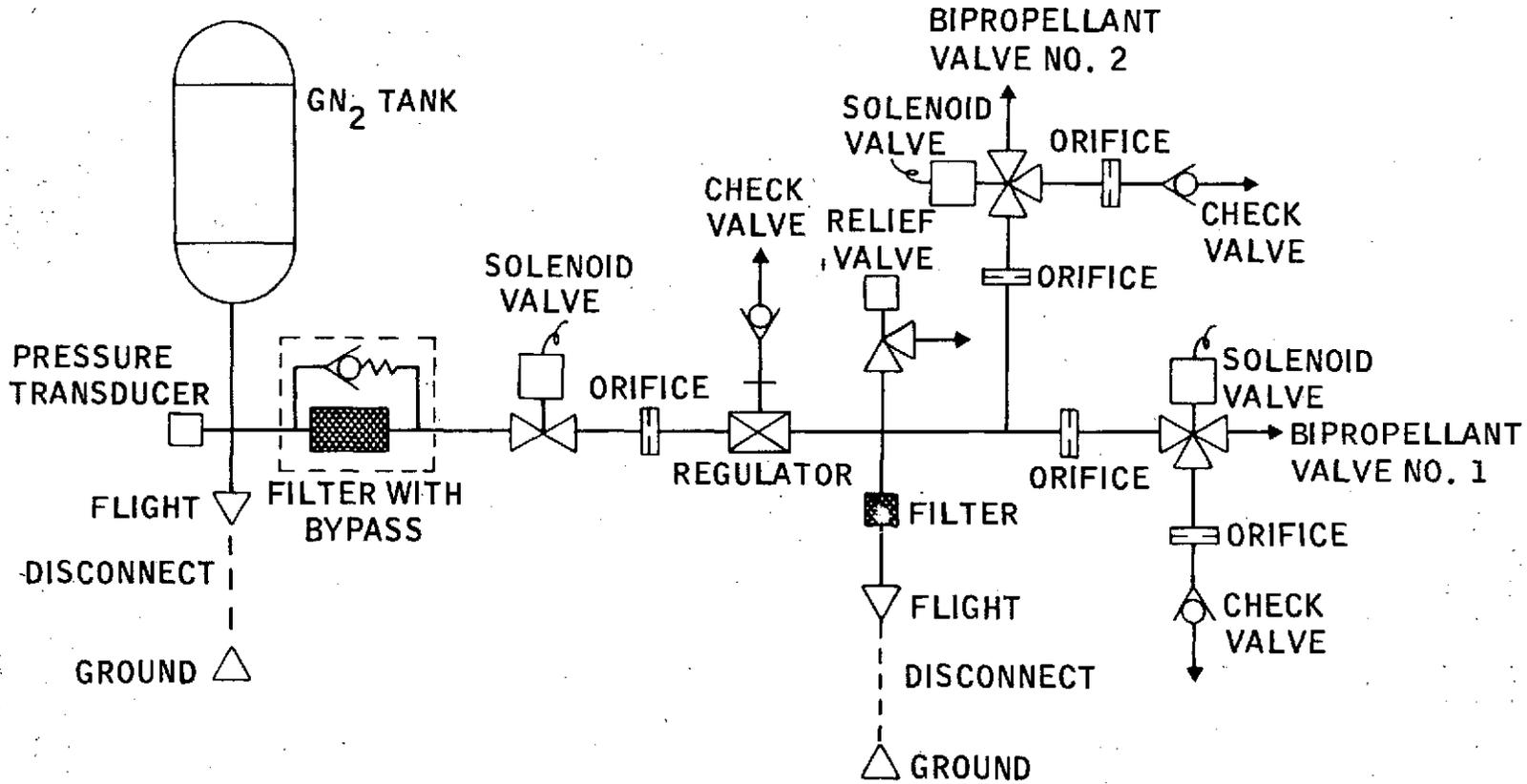


Figure 4-2

PNEUMATIC SUPPLY SYSTEM SCHEMATIC

the mission. In addition to these key components the pneumatic supply system includes a relief valve, two quick disconnects for filling the system and for direct operation of a ground supply, a pressure transducer for monitoring GN₂ tank pressure and several orifices, filters, and check valves to control control the GN₂ flow.

In sizing the pneumatic supply tank the assumptions listed in Table 4-IV were made.

TABLE 4-IV

PNEUMATIC SUPPLY TANK SIZING CRITERIA

Mission Duration	30 Days
Assumed Leakage Rate	250 SCC per hour
Tank Storage Pressure	2000 psi
Minimum Regulator Inlet Pressure	500 psi
Actuator Operating Pressure	250 psi
Number of Actuators/Mission	20
Material of Construction	Stainless Steel
Storage Margin	2 to 1

To determine weight and power requirements of the pneumatic supply package it was assumed that off the shelf components available from other aerospace programs would be utilized. Based on information available in Marquardt's component files and on contacts with a number of component suppliers a weight estimate was prepared for the pneumatic supply system. This weight estimate by component is presented in Table 4-V. Accordingly a total pneumatic supply system weight of 11 pounds was determined. Based on this determination it became apparent that all valve and actuation system concepts utilizing pneumatic actuation would incur a heavy weight penalty for the tradeoff study. Furthermore it was determined that an additional 112 watts of power would be required per pneumatic supply package to operate the three valves and pressure transducer. A study of the pneumatic supply package also indicated that the goal of zero maintenance throughout the life of the space shuttle could not be achieved with the pneumatic supply package in that indeed a number of functional checks and the filling of the pneumatic supply package with gaseous nitrogen had to be accomplished prior to each space shuttle flight. Table 4-VI is a list of typical maintenance requirements for the pneumatic supply system. These maintenance requirements in combination with the weight penalty and the unreliability penalty inherent in the pneumatic supply package because of the significant number of components placed a heavy penalty upon electropneumatically operated valves during the preliminary valve tradeoff study.

4.1.4 VALVE DESCRIPTIONS

During the preliminary valve tradeoff study a number of preliminary valve designs were prepared. These designs served the primary purpose of obtaining valve weight estimates to permit a comparison of potential valve concepts on a weight basis. These concepts also served as the basis for the list of preliminary valve tradeoff study

TABLE 4-V

PNEUMATIC SUPPLY SYSTEM WEIGHT BREAKDOWN

QTY.	TYPE	WEIGHT (LBS.)
1	GN ₂ TANK	2.2
2	GN ₂ FILL AND DRAIN	0.3
1	FILTER WITH BYPASS	0.7
1	PRESSURE TRANSDUCER	0.6
1	PREVALVE	0.9
5	ORIFICES	0.2
1	REGULATOR	1.5
1	CHECK VALVE AND TEST PORT	0.3
1	RELIEF VALVE	0.6
1	FILTER	0.1
2	VENT PORT CHECK VALVES	0.3
2	3-WAY SOLENOID VALVES	1.8
1	MOUNTING PANEL AND CLAMPS	1.0
	MISCELLANEOUS PLUMBING	0.5
	TOTAL WEIGHT	11.0

TABLE 4-VI

PNEUMATIC SUPPLY SYSTEM MAINTENANCE REQUIREMENTS

PREFLIGHT:

1. CHARGE GN₂ SYSTEM - VERIFY GN₂ PRESSURE
2. OPEN PRE-VALVE
3. VERIFY REGULATED OUTLET PRESSURE
4. LEAK CHECK SOLENOID VALVE NO. 1 IN CLOSED POSITION
5. LEAK CHECK SOLENOID VALVE NO. 2 IN CLOSED POSITION
6. ENERGIZE SOLENOID VALVES
7. LEAK CHECK SOLENOID VALVE NO. 1 IN OPEN POSITION
8. LEAK CHECK SOLENOID VALVE NO. 2 IN OPEN POSITION
9. CLOSE PRE-VALVE
10. LEAK CHECK PRE-VALVE/REGULATOR COMBINATION
11. RECHARGE GN₂ SYSTEM - VERIFY GN₂ PRESSURE

IN FLIGHT:

1. MONITOR GN₂ TANK PRESSURE

POST FLIGHT:

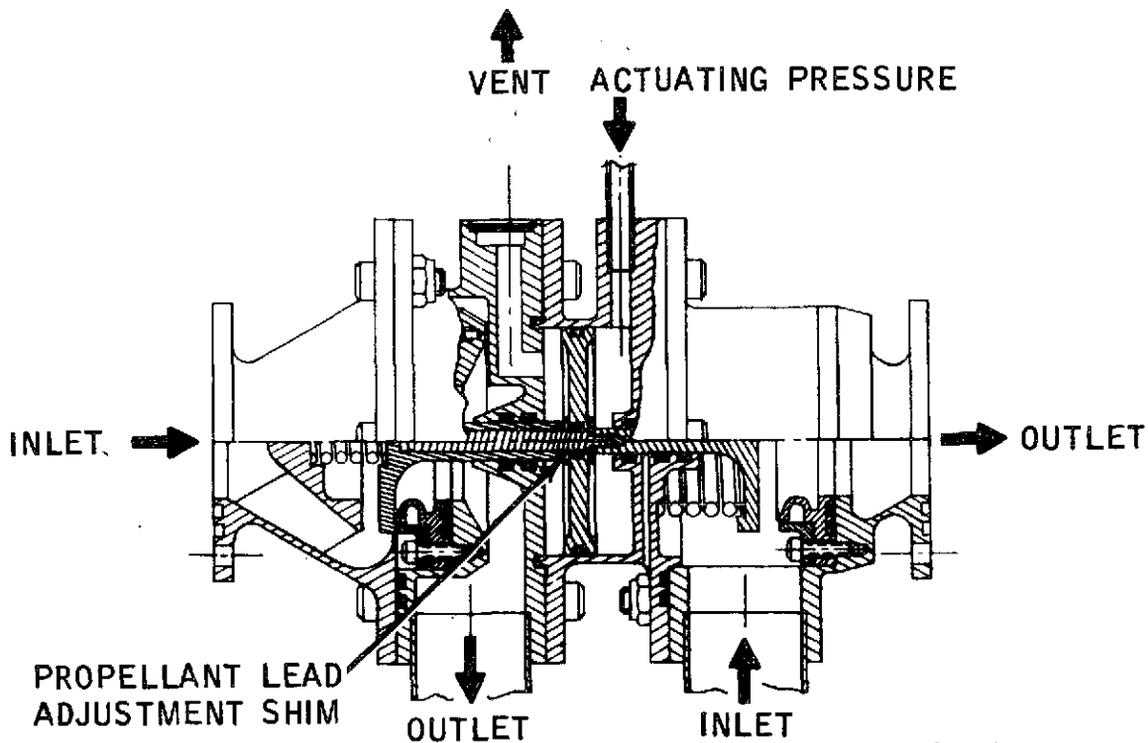
1. REPEAT PRE-FLIGHT CHECKOUTS EXCEPT FOR GN₂ SYSTEM CHARGING

NOTE: IF THE CHECKOUTS REVEAL A FAILURE, THE WHOLE PNEUMATIC SUPPLY SYSTEM IS REMOVED.

candidates presented in the next section. Each of the preliminary valve designs prepared will be briefly described in the following paragraphs.

In-Line Pneumatically Operated Poppet Valve

A cross section of the bipropellant valve configuration is shown in Figure 4-3. The pneumatic supply system required for operation of this valve was discussed in the preceding Section. This particular bipropellant valve does include the capability for adjusting the lead of one of the propellants. This adjustment may be made by simply replacing an adjustment shim which spaces the relative position between the pneumatic piston actuator and the lagging valve. The valve features all titanium (6AL-4V) construction with the exception of the return springs, the poppet and shaft assemblies, static and dynamic seals, the seat and the poppet stop, and the nuts and bolts required to make the assembly. The poppets, seat, and poppet stop are made from Inco 718. In the configuration shown, the seat is a gold-plated concept which was previously demonstrated by The Marquardt Company for over 100,000 cycles of actuation in gaseous nitrogen. The return springs are made from Inco X-750 and the static and sliding seals are of the Teflon-jacketed configuration. Sliding surfaces adjacent to the poppet shaft and at the pneumatic piston are impregnated with Teflon utilizing the Canadizing process.



IN-LINE PNEUMATIC OPERATED POPPET VALVE

The valve has been designed to permit disassembly of and access to all of its components. Valve operation is achieved by pressurizing the right side of the pneumatic piston. The pneumatic force thus generated will first lift off the poppet in the right half of the valve and then pick up the poppet in the left valve to open it as well. In Figure 4-3, the valve travel is at an intermediate position wherein the right poppet is partially open and the left poppet has not yet started to open. When the pneumatic pressure on the right side of the piston is vented overboard, the coil springs in each of the propellant valves return the poppets to the closed position.

Parallel Poppet Pneumatically Operated

As the name implies, the parallel poppet bipropellant valve features (see Figure 4-4) two poppets arranged in parallel and connected through a common yoke to the pneumatic piston actuator located between them. This particular valve concept is extremely simple and highly reliable. All three dynamic seals are hydroformed bellows to assure zero propellant and pneumatic leakage. To operate the valve, the pneumatic piston is pressurized with gas from the pneumatic supply system previously discussed in Section 4.1.3 and shown in Figure 4-2. Pneumatic operating pressure is 250 psia. To close the valve, the pneumatic pressure is vented overboard and the combined spring rates of the three bellows and three sets of axial guidance flexures close the valve. The common yoke has been sized so as to feature sufficient "give" to make certain that both of the poppets will seat effectively.

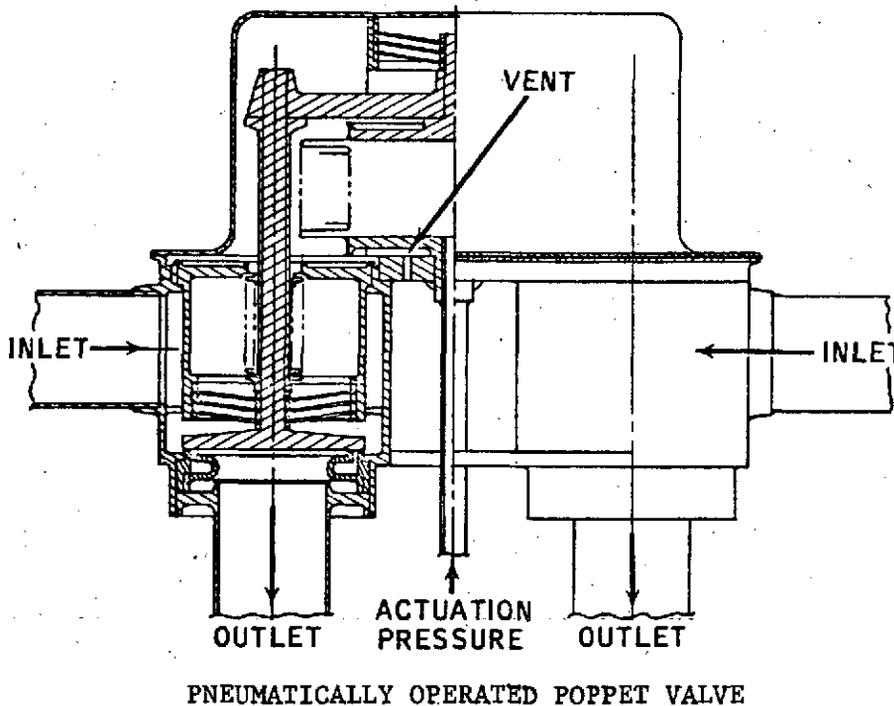


Figure 4-4

The valve concept is completely friction-free since it features axial guidance flexures for positioning the poppets as well as the actuator and the dynamic seals are bellows. This feature should make the valve completely maintenance-free and therefore permits the all-welded construction shown. The poppet/seat interface as shown consists of a flat bottom poppet and

a gold-plated lip seal in combination with a poppet stop. The valve is made entirely of Inco 718.

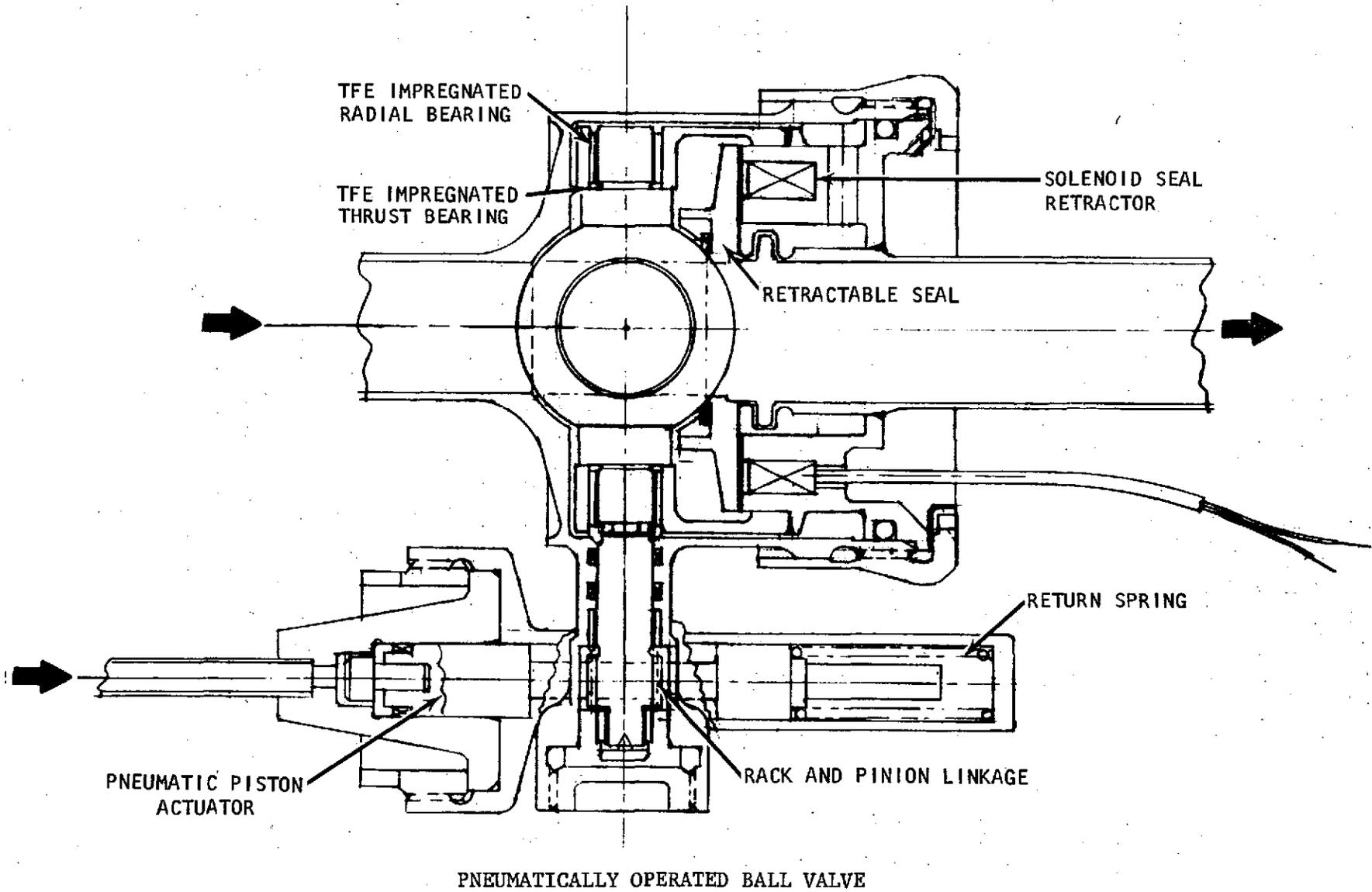
Pneumatically-Operated Ball Valve With a Solenoid Retracted Seal

The subject valve is shown in Figure 4-5. The pneumatically-actuated ball valve is actually a bipropellant valve with the actuator rack driving two ball shafts simultaneously. The second ball has been eliminated in this view for clarity. The basic design incorporates a trunion-mounted ball spherically lapped on its sealing surface and accurately positioned to maintain alignment with the retracting valve seat carrier. The possibility of various types of salts being deposited immediately downstream of the sealing closure interface makes it mandatory that the seal be fully retractable as opposed to allowing the ball to slide across the seal during the opening motion. Except for the seal retraction solenoid, the actuator return spring, and the Teflon seal, the valve is made entirely of titanium (6AL-4V). The excellent material compatibility, coupled with its low weight, make this material ideal for the OME shutoff valve application. Thrust and radial bearings at the ball shaft, as well as the rack and pinion gear interface of the actuator and the sliding pneumatic piston, are treated with a PTFE infusion process called Canadizing. The process infuses PTFE (Polytetrafluoroethylene) into oxide or nickel chrome porosities to produce a permanent lubricity to the surface with a coefficient of friction of 0.05. Galling and seizing are eliminated and wear is minimized. The surface is hard (about 60 to 65 Rc) and can be machined by honing with carbide tools. This process has been applied and successfully used in the Apollo program Moon Probe Drive System, as well as in the LEM ascent engine.

The seat configuration incorporated in the ball valve is the trapped Teflon concept. The seal is retracted a distance of 0.005 inch prior to the initiation of ball movement by means of the seal retraction solenoid. This solenoid is constructed of Type 304L and 446 stainless steel, both of which are fully compatible with the OME operating fluids and environment. The retraction solenoid consumes 14 watts of electrical power.

4-13

Figure 4-5



The solenoid is energized simultaneously with the signal to operate the ball valve. Since the solenoid response is much faster than that of the pneumatic actuator or even that of an electrical torque motor, the seal retracts before any ball motion occurs. Once the ball valve leaves the closed position, the solenoid is also provided with electrical power from the ball valve closed position microswitch. Consequently, during the closing motion of the ball valve, the solenoid remains energized until the closed position microswitch is again triggered, thus assuring that the ball has reached the fully closed position before the seal is reapplied.

A pneumatic supply system, as described in Section 4.1.3, and as shown schematically in Figure 4-6, is required to operate this actuator. For the opening motion, the piston is pressurized and for the closing motion the piston is vented and the piston return spring causes the valve to move to the closed position. Redundant Teflon-jacketed seals prevent propellant leakage from the ball valve out around the ball valve shaft. The position indicator is not shown in this view but is located in the actuator spring cavity. The pneumatic actuator also utilizes Teflon-jacketed seals. Extra flow passages have been provided around the solenoid actuator to enhance purge gas circulation during decontamination.

It should be noted that although Figure 4-5 shows pneumatic actuation, electric torque motor actuation can just as easily be accomplished and is, according to the preliminary trade-off, substantially lighter. Either direct torque motor drive or torque motor drive through gears such as described during subsequent discussions of the dual poppet valve which is shown in Figure 4-6 can be readily incorporated.

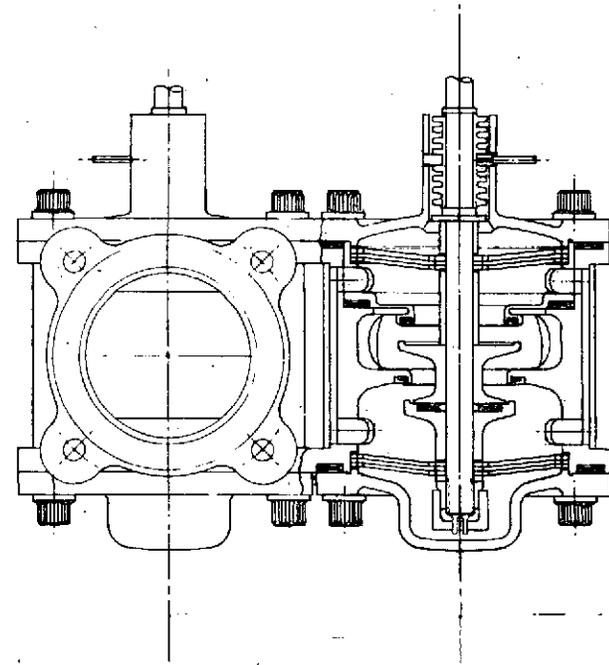
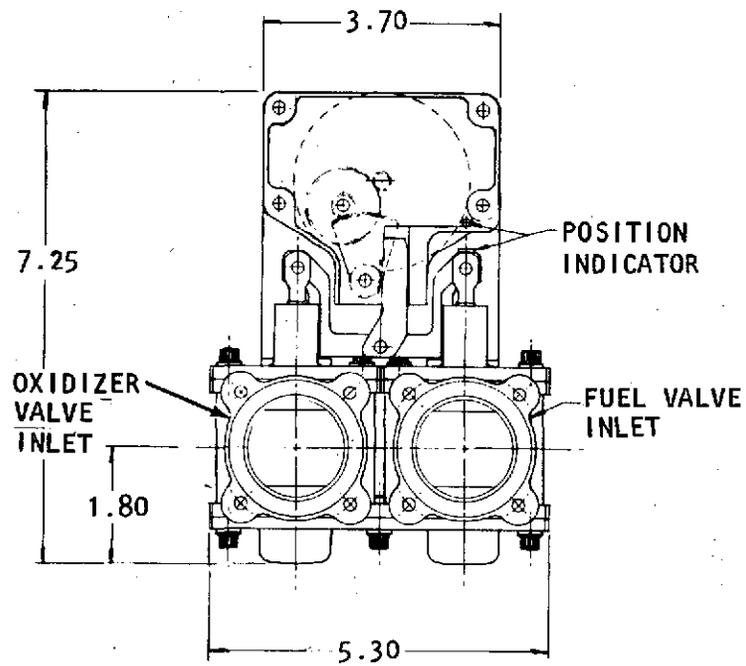
Ball Valve, Torque Motor Actuated

The torque motor actuated ball valve is shown in Figure 4-7. This valve utilizes a single torque motor to simultaneously drive two balls. With the exception of the torque motor actuator and the mechanically retracted seal, the valve is quite similar to the pneumatically-actuated ball valve just discussed and presented in Figure 4-5. Mechanical retraction of the valve seat is accomplished by a ring cam which is driven by the valve torque motor through reduction miter gears to raise the valve seat before opening ball rotation begins. Seating occurs only when the ball is in the closed position.

Torque Motor-Operated Dual Poppet Pressure-Balanced Bipropellant Valve

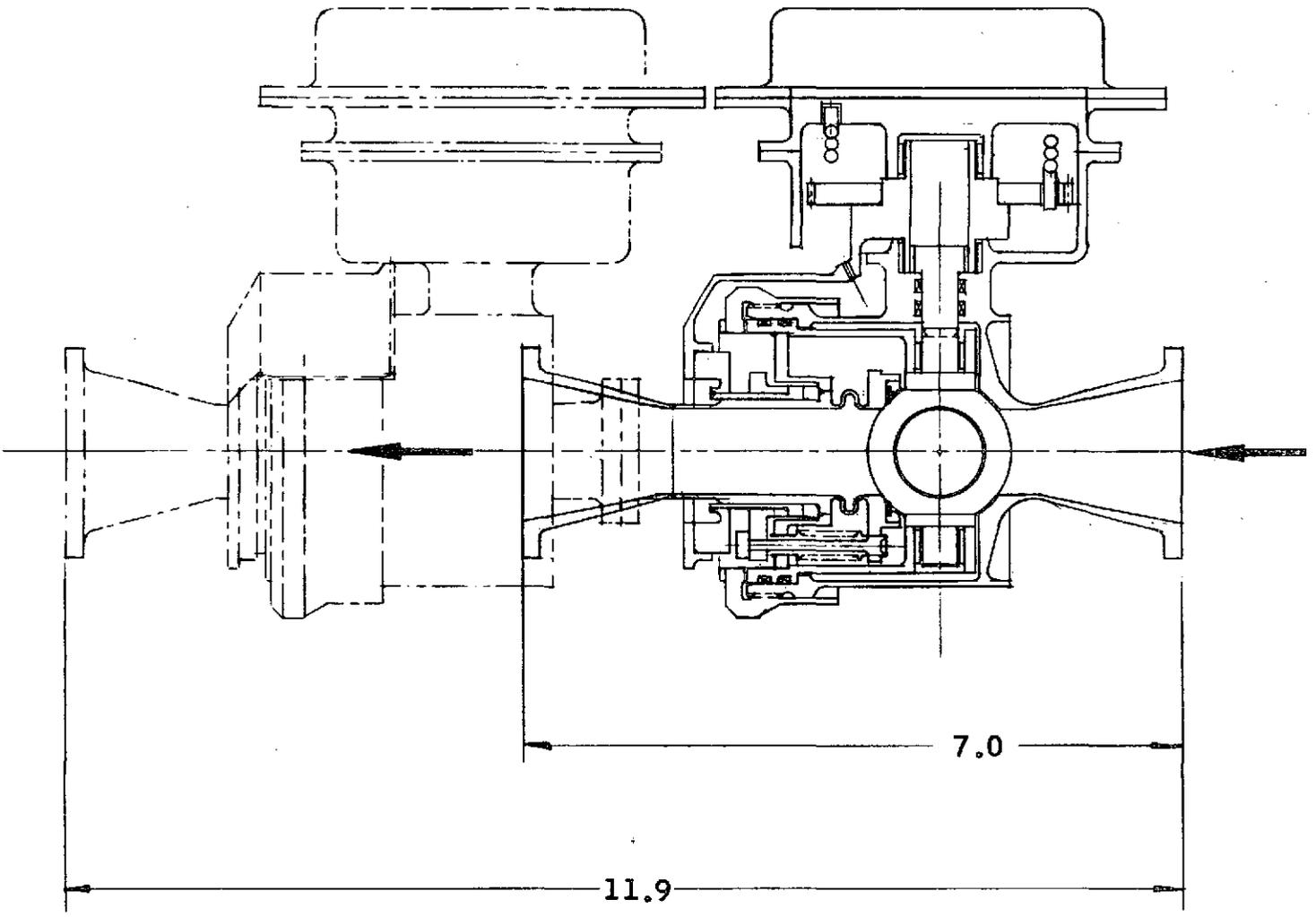
The subject bipropellant valve is shown in Figure 4-6. The dual poppet concept has been used extensively by The Marquardt Company in mass flow regulators for ramjet engines. Its adaptation to the OME shutoff valve offers several very noticeable advantages. Dual poppet pressure balancing minimizes actuator force requirements since it eliminates the pressure unbalance forces normally encountered in other poppet valves. The only

4-15



TORQUE MOTOR OPERATED POPPET VALVE

Figure 4-6



TORQUE MOTOR OPERATED BALL VALVE

4-16

Figure 4-7

actuation force required is that needed to overcome the spring forces of the axial guidance flexures and the bellows dynamic seal which are sized to provide the necessary sealing closure interface forces to achieve the leakage requirement. The dual poppet pressure balancing technique does away with the extra bellows or dynamic seal normally required for more conventional pressure balanced poppet valves. The dual poppet concept is a very simple concept and has the capability of sealing nearly as reliably as a single poppet valve since the total seal area of a single poppet valve is nearly identical to the combined sealing areas of the dual poppets. Simultaneous sealing of both poppets in the dual poppet concept is assured by making the carrier of one of the seats a spring element, such that it can be deflected approximately 0.003 to 0.005 inch and will achieve an effective seal with the poppet at any position within that dimensional range. The concept of a spring-loaded seat has been demonstrated extensively by The Marquardt Company in support of high cycle life valves (over one million cycles) for the NASA-Lewis Research Center.

The poppet seat interface concept shown in Figure 4-6 is again the trapped Teflon discussed previously. The poppets are fully flexure-guided and the shaft seal features a hydroformed redundant bellows assembly. Thus, frictional forces have been eliminated to achieve a highly reliable design. Materials of construction are titanium for all parts except the poppets, poppet shaft, static seals, bellows shaft seal, and the closure plate that the bellows shaft seal is welded to. These latter parts, as well as the axial guidance flexures, are made from Inconel 718.

Utilization of the dual poppet concept results in a very attractive valve package. Since the two poppets are smaller in diameter than a single poppet would be, the overall diameter of the poppet housing is reduced. Furthermore, because of the dual flow path, a straight through flow arrangement is desirable and this arrangement is most suitable for series and quad redundant valve packaging.

Operation of the dual poppet bipropellant valve may be accomplished with pneumatic or torque motor actuators. Figure 4-6 shows a dual poppet bipropellant valve utilizing a brushless DC torque motor and gear linkage for each bipropellant valve. The torque motor gear train shown in this figure has a ratio of 33:1. Three revolutions of the torque motor result in a 32.6 degree rotation of the poppet actuation pinion. Relatively low actuation forces allow the gear train to be made of titanium and to be lubricated by means of the Canadizing process discussed previously.

The torque motor is connected to the two poppets by means of a yoke which has been stressed so as to permit limited deflection of either end to assure simultaneous seating of both halves of the bipropellant valve. Valve operation is achieved by simply energizing the torque motor. To close the valve, the torque motor is deenergized and the flexure and bellows spring forces return the poppets to the seating position.

4.1.5 CONCLUSIONS

Based on the preliminary valve design concepts presented in the preceding section and the study of possible combinations of actuators and shut off devices featured in these valves the six most promising valve configurations were compared. A summary of this comparison is presented in Table 4-VII. The comparison was made on the basis of weight, reliability, electrical power, and maintenance requirements. Table 4-VII also lists opening and closing responses. The valves were sized to feature the same opening response such that a valid comparison of electrical power and weight could be made. The only valve that did not feature the 250 millisecond opening response was the solenoid poppet valve since the solenoid inherently has faster response characteristics. The closing response is listed for information only and was used primarily to verify that the particular valve had an acceptable closing response. Determination of a valve and actuation weight was based on 3 factors. First a weight penalty was assigned to penalize those concepts that required more electrical power for operation since these concepts would also require heavier electrical supply wiring in the space shuttle. Consequently the weight values listed under the heading electrical supply are directly proportional to the electrical power required by the particular valve and actuation system. The column entitled pneumatic supply weight assesses a weight of 22 pounds to all those valve actuation system concepts that employ pneumatic power and that therefore require two of the pneumatic supply packages as described in Section 4.1.3 for each quadredundant valve package. The third column entitled quad valve weight then lists the actual weight of the four bipropellant valves including actuators. Consequently to obtain the total weight assessed each valve and actuation system concept the three columns entitled electrical supply weight, pneumatic supply weight, and quad valve weight must be added together. When these columns are added together it is evident that the pneumatically operated valve concepts are substantially heavier than the electric torque motor operated valve concepts.

Reliability and maintenance requirements were evaluated on the basis of 10 points maximum for the best configuration. Here again a comparison of the pneumatically operated valve concepts with the electric torque motor operated concepts resulted in the conclusion that the pneumatically operated concepts are substantially less reliable and require substantially more maintenance than the electric torque motor concepts. Accordingly low ratings were assigned the pneumatically operated concepts in both of these categories. Finally a comparison of the electric power requirements of the six valves disclosed that they range between a minimum of 120 watts to a maximum of 360 watts which was considered an acceptable range and which therefore did not affect the preliminary valve tradeoff.

Based on the data presented in Table 4-VII it was concluded that the two electric torque operated concepts (the dual poppet valve and the ball valve with a retracted seat) were the most promising concepts for the OME valve and actuation system application. Not only were they more than 30% lighter than the nearest pneumatically operated concept but they were also considered substantially superior from the point of view of reliability and maintenance requirements.

TABLE 4-VII

PRELIMINARY VALVE TRADE OFF SUMMARY
COMPLETE QUADVALVE AND ACTUATION SYSTEM

VALVE TYPE	ACTUATOR TYPE	WEIGHT (LBS.)			RELI ABILITY (10 PTS. MAX.)	ELECT POWER (WATTS)	CLOSING RESPONSE (MS)	OPENING RESPONSE (MS)	MAINTENANCE REQUIREMENTS (10 PTS. MAX.)
		ELECT SUPPLY	PNEUM SUPPLY	QUAD VALVE					
DUAL POPPET (BALANCED)	ELECTRIC TORQUE MOTOR	4.0	0	17.0	9	224	210	250	10
DUAL POPPET (BALANCED)	PNEUMATIC PISTON	4.0	22.0	16.8	5	224	250	250	6
POPPET (BALANCED BY BELLOWS)	SOLENOID	6.0	0	EXCESSIVE	7	360	50	100	10
POPPET (UNBALANCED)	PNEUMATIC	4.0	22.0	14.8	6	224	250	250	6
BALL (RETRACTED SEAT)	PNEUMATIC	4.0-6.0	22.0	22.6	4	224-360	250	250	5
BALL (RETRACTED SEAT)	ELECTRIC TORQUE MOTOR	2.8-4.0	0	23.4	8	120-232	160	250	9

4-19

4.2 FINAL VALVE TRADEOFF

The purpose of the final valve tradeoff study was to perform detailed analysis of the two most promising concepts namely the electric torque motor operated ball valve and the electric torque motor operated dual poppet valve. In the case of the ball valve two means for retracting the seal had been investigated during the preliminary valve tradeoff. These were retraction by a solenoid actuator and external mechanical retraction. The mechanical retraction method was called external because the gears and the cams require to transmit the motion from the torque motor to the seal were located external to the propellant cavity of the valve and this propellant cavity was breached only by three pull rods which were sealed at the valve housing by means of bellows. Since the external mechanical retraction technique was somewhat complex it was decided in conjunction with the NASA technical monitor to prepare one additional ball seal retraction design which would feature internal mechanical retraction. In other words in this design the gears and cams required to transmit motion from the torque motor to the seal were located inside the propellant cavity. The internal mechanical retraction design appeared to offer a simpler more compact overall configuration. Thus a final valve tradeoff was made involving essentially four valve designs. One of these was the dual poppet valve and the other three were the ball valve with three different techniques for retracting the seal. These valve candidates as well as the analysis performed to size these valves are presented in this section of the report.

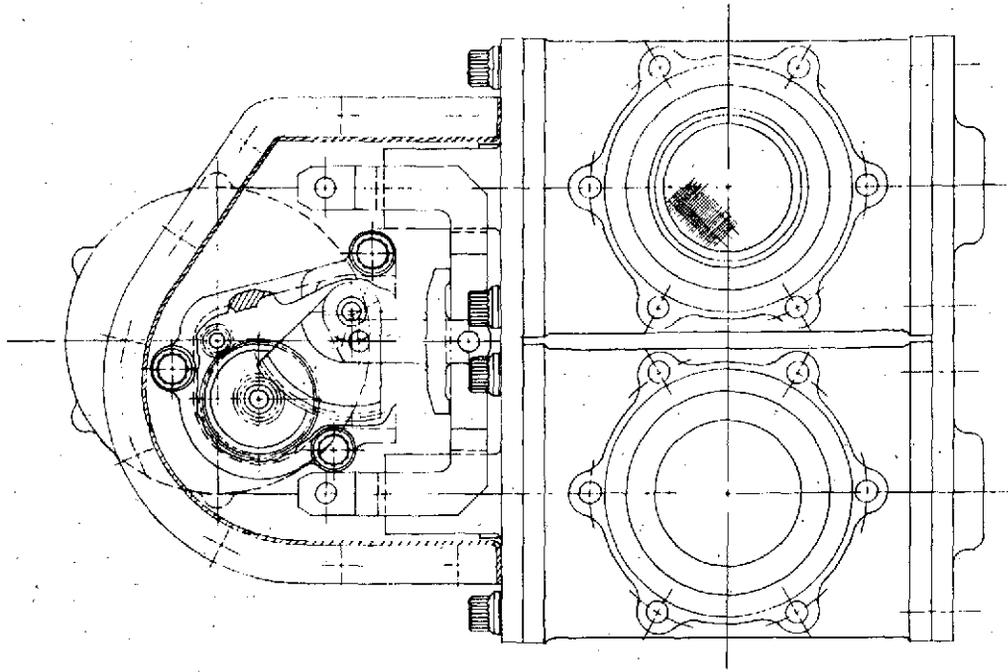
4.2.1 DESCRIPTION OF FINAL TRADEOFF CANDIDATES

Torque Motor Operated Dual Poppet Valve

The torque operated dual poppet valve was previously described in section 4.1.4 and the preliminary design of this valve is shown in Figure 4-6. Based on the analysis performed during the final tradeoff study this design was updated and the final design configuration considered is presented in Figure 4-8. Details of the valve mechanism and of the individual poppet valve are not shown in Figure 4-8 since they remain essentially the same as those shown in Figure 4-6. The final dual poppet valve includes an inlet screen on the fuel side (right side of Figure 4-8). The valve is designed for a counter flow arrangement such that the valve flange located on the left side is the outlet of the oxidizer valve. The valve actuator was refined to include an improved linkage arrangement and correctly sized gears. In addition the actuator housing was redesigned to provide a location for the electronic package which is required for operation of the brushless DC torque motor. Operational characteristics and construction features remain the same as discussed in Section 4.1.4.

Ball Valve - External Gear Seal Lift-Off

The torque motor operated ball valve featuring an external mechanical seal retraction design was previously discussed in Section 4.1.4 and shown in Figure 4-7. Further analysis of this concept did not significantly change the valve configuration except



DUAL POPPET VALVE

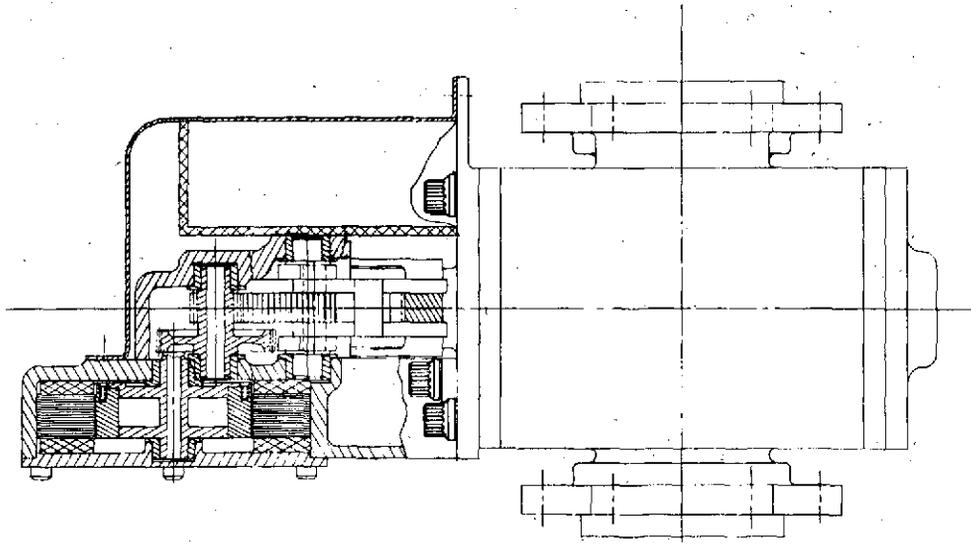


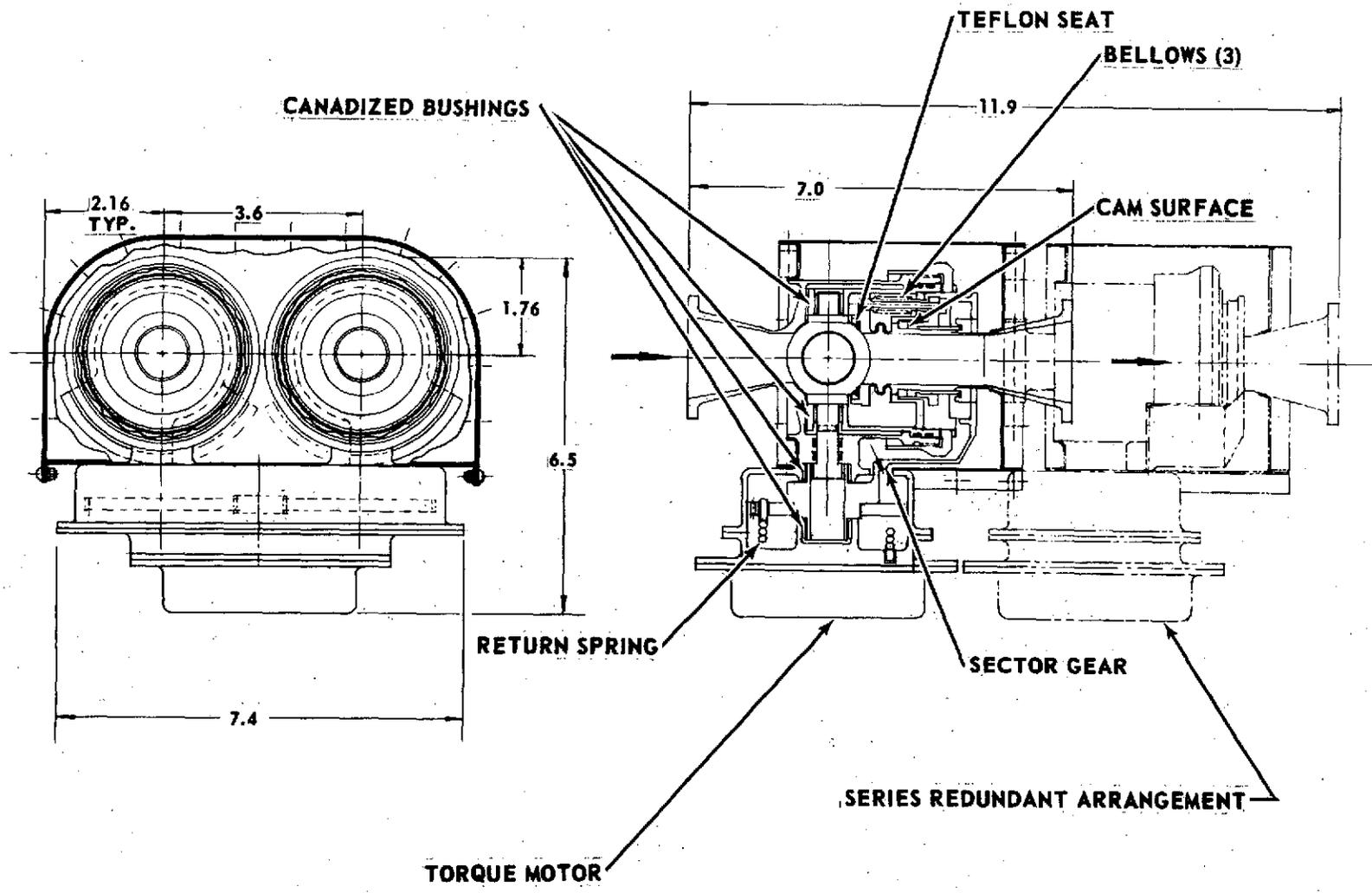
Figure 4-8

that it was considered necessary to add a housing around the externally located retraction mechanism to prevent the introduction of contaminants into the mechanism. This modification is shown in Figure 4-9. Figure 4-9 also presents a side view of the bipropellant valve and specifies valve envelope dimensions. Operational characteristics and construction features are as described in Section 4.1.4.

Ball Valve - Internally Actuated Seal Lift-Off

This particular valve concept is shown in Figure 4-10. The torque motor actuator of this valve is similar to that shown in Figure 4-9 except that the drive required to lift off the seal is now located inside the propellant cavity. Seal liftoff is accomplished off a sector gear located on the same axis as the ball valve shaft. This sector gear drives a ring gear which is located concentric with the valve flow axis and therefore concentric with the valve seal. As the ring gear is turned a cam surface on the ring gear lifts off the valve seal actuating ring and thereby lifts off the valve seal. The seal liftoff motion is illustrated in Figure 4-11. One of the features of this lift off mechanism is that it is required to lift off the seal before any rotation of the ball occurs during the opening motion. Similarly during the closing motion the ball must be returned to the closed position before the seal is allowed to extend and come in contact with the ball. These requirements are necessary to assure that there will be no scrubbing of the seal during the opening or closing motion.

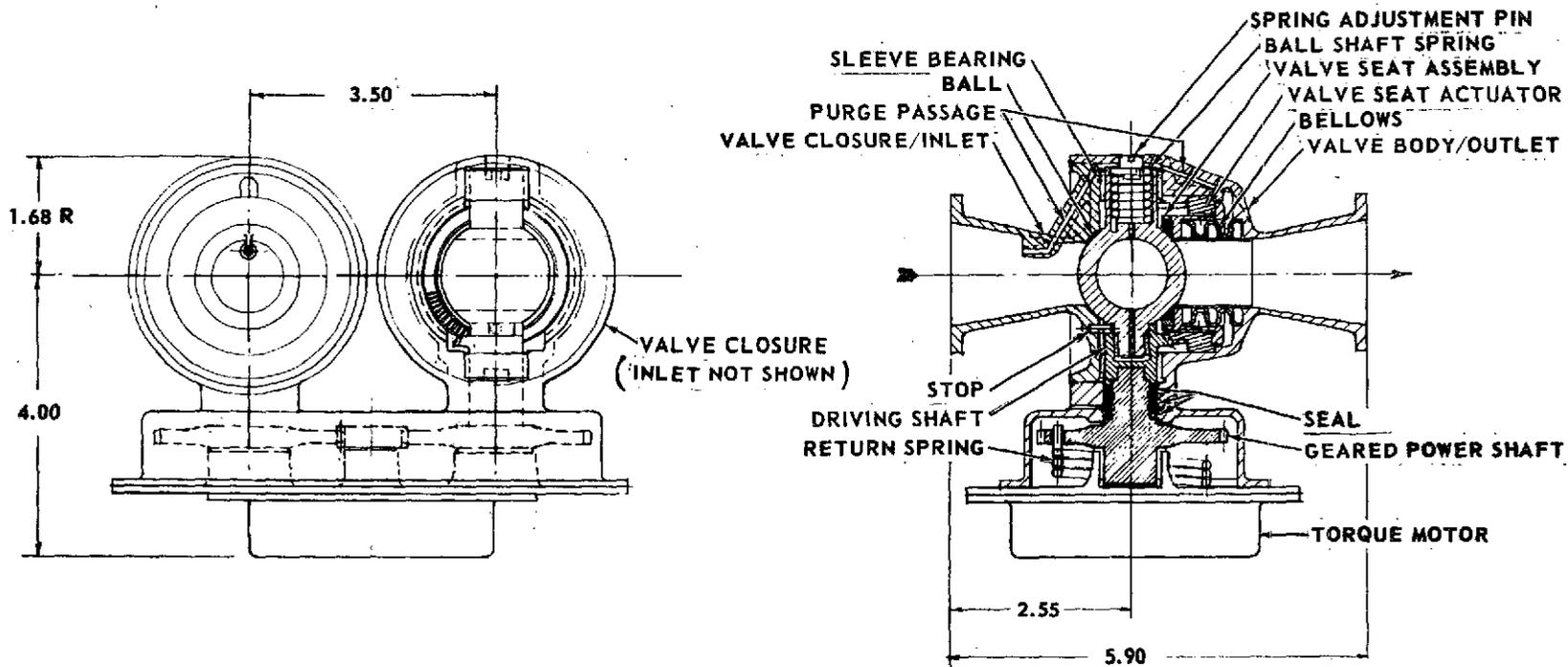
In Figure 4-11 the cross lined portion of the shaft is attached to the ball and also to the return spring. The unlined portion of the shaft as well as the sector gear are attached to the torque motor. Following the sequence from diagrams 1-5 the lift off action may be observed. In Diagram 1 the valve and seal are closed and a gap of approximately 20° of arc exists between a motor shaft and the ball shaft. By driving the torque motor the ring gear and therefore the seal lift off mechanism are initiated immediately as shown in Diagram 2. The ball valve shaft motion does not start until the 20° of arc have been accomplished by the torque motor shaft and the torque motor shaft is in effect striking the mating surface on a ball shaft. At that point both the ball shaft and the torque motor shaft turn; however at that point also the seal has already lifted off as a result of the cam action on the ring gear and there is no further motion of the seal. Diagram 3 shows the valve reaching the fully opened position when the ball shaft strikes the open stop. In Diagram 4 the valve is returning to the closed position as a result of the torsion spring driving the ball valve shaft and this shaft in turn driving the torque motor shaft. Diagram 5 shows the ball valve shaft striking the stop in the closed position. At this time the momentum of the torque motor, sector gear, and ring gear as well as the cam action of the seal wanting to move up against the ball as a result of seal bellows preload cause the torque motor, sector gear, and ring gear to continue the rotary motion until the torque motor shaft strikes the ball shaft to eliminate the 20° arc lead of the ball shaft. This then returns the seal to the closed position as shown in diagram 1.



BALL VALVE - EXTERNAL GEAR SEAL LIFT-OFF

4-23

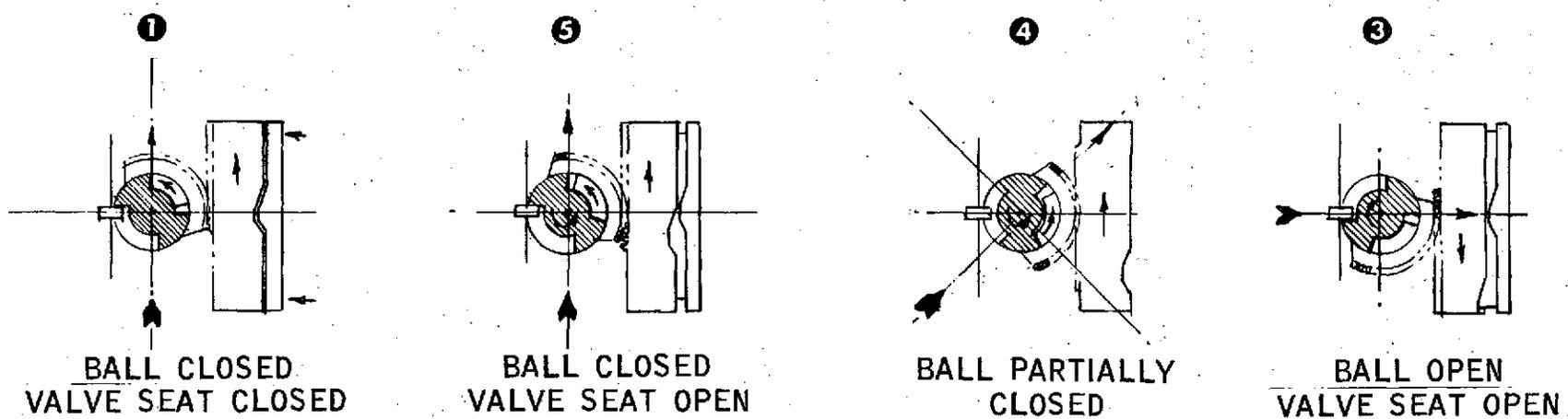
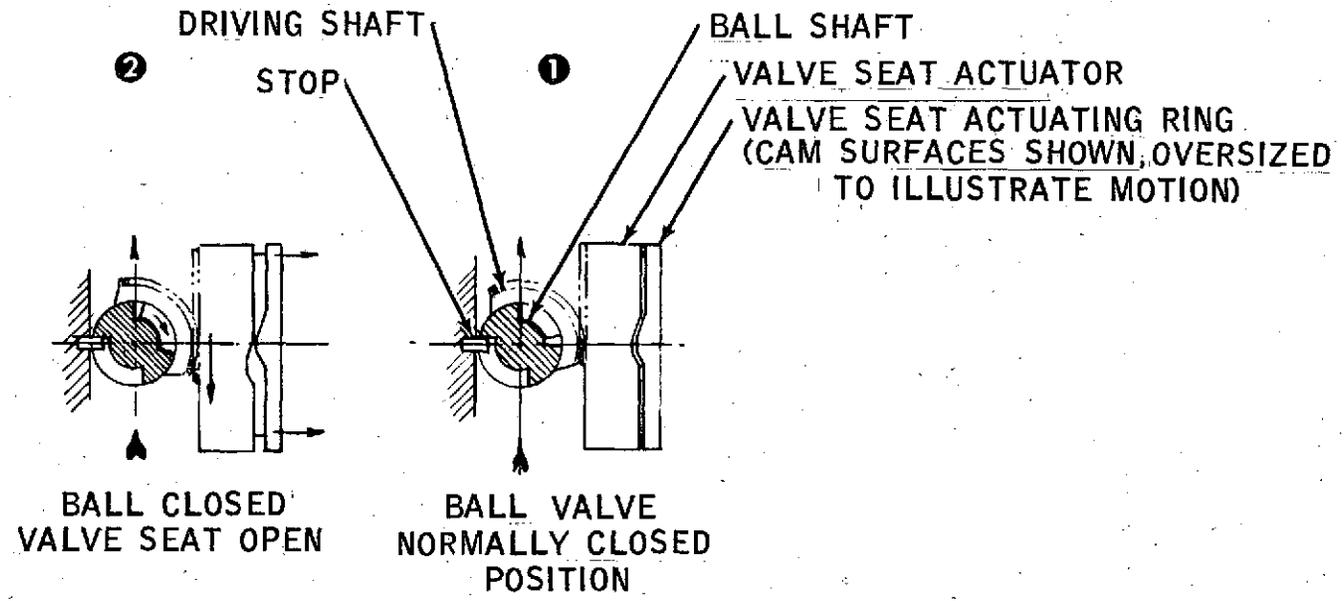
Figure 4-9



BALL VALVE - INTERNALLY ACTUATED SEAL LIFT-OFF

4-24

Figure 4-10



SEAL LIFT-OFF MECHANISM

4-25

Figure 4-11

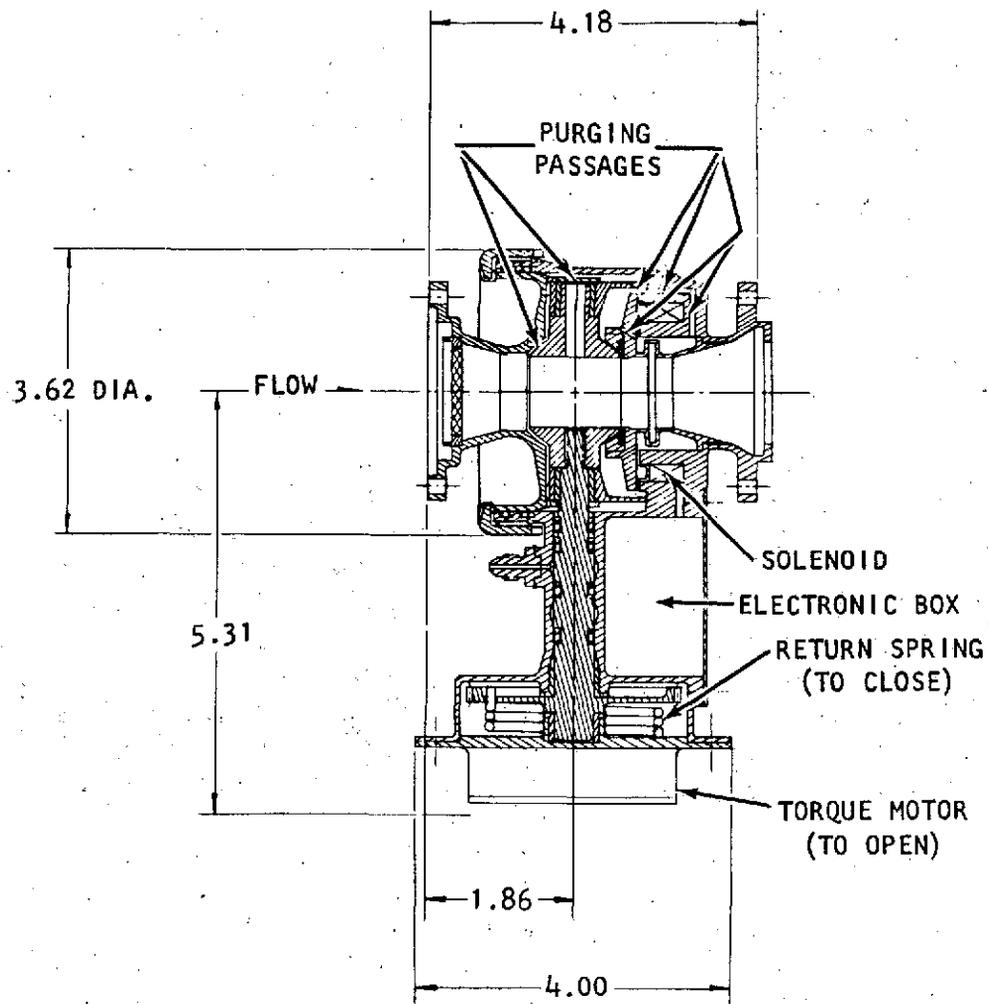
Figure 4-10 also shows a purge passage which picks up the total pressure of the incoming purge gas and routes it towards the valve housing to purge out relatively dead cavities around the ball bushings and in the seal liftoff mechanism. This same principal of utilizing total pressure to achieve purge gas flow within the valve housing was subsequently incorporated in all the other valve designs as well. Operational characteristics and construction features of this valve are essentially the same as those of the external gear seal liftoff ball valve with the exception of the liftoff mechanism.

Ball Valve With Solenoid Retracted Seal

This particular valve concept is presented in Figure 4-12. The solenoid retracted seal concept was previously discussed in Section 4.1.4 and presented in Figure 4-5 except that Figure 4-5 featured a pneumatic actuator for the ball valve whereas the final valve tradeoff was made with the configuration shown in Figure 4-12 which features a torque motor actuator. The ball valve shown in this latter figure has also been refined to include an inlet screen as well as to include a purge passage for clearing out propellant residues from dead cavities inside the valve. Figure 4-12 also shown the redundant shaft seal arrangement that was subsequently employed in the ball valve design. This arrangement resulted from comments by Rockwell International Space Division technical personnel who expressed a desire to have redundant shaft seals followed by a third seal to permit leak check of the redundant shaft seals. In addition Marquardt technical personnel decided to include a fourth seal which would prevent contamination from entering the torque motor housing. Operational characteristics and construction details are similar to those discussed for the other ball valves.

4.2.2 VALVE ANALYSIS APPROACH

The valve analysis approach pursued during the sizing and the determination of the performance characteristics of the final valve tradeoff candidates is presented schematically in Figure 4-13. The sizing of the valve is initiated with the determination of the flow area to meet the valve flow and pressure drop requirements. This determination includes an allocation of quad valve pressure drop with respect to each valve and each filter and in the case of the poppet valve a tradeoff of seat diameter versus stroke. Once the flow area is determined the force or torque required to operate the valve mechanism is calculated. This calculation takes into consideration any force reduction means, friction characteristics, force variations as a function of stroke, as well as the required sealing forces. The sealing forces are of course derived from the sealing closure interface configuration and the allowable leakage requirements. After the sealing closure force or torque characteristics are determined an actuator can be sized. In sizing the actuator force or torque reduction means as well as frictional characteristics and thermal or voltage degradations are considered. Upon completion of the actuator sizing the response characteristics of the valve as well as the power requirements to achieve the desired response characteristics are calculated.



BALL VALVE WITH SOLENOID RETRACTED SEAL

VALVE ANALYSIS APPROACH

FLOW AREA DETERMINATION
 A. QUADVALVE ΔP ALLOCATION
 B. C_D DATA
 C. SEAT DIA. vs STROKE TRADE OFF

LIFT OFF FORCE/TORQUE DETERMINATION
 A. FORCE REDUCTION MEANS
 B. FRICTION CHARACTERISTICS
 C. SEALING FORCES
 D. DYNAMIC FORCES
 E. FORCE vs STROKE

ACTUATOR SIZING
 A. FORCE/TORQUE REDUCTION MEANS
 B. FRICTION CHARACTERISTICS
 C. THERMAL EFFECTS
 D. NET TORQUE REQUIRED

RESPONSE DETERMINATION
 A. INERTIA/LIFT OFF TORQUE/ACTUATOR TORQUE
 B. OPENING vs CLOSING RESPONSE
 C. POWER
 D. FORCE REDUCTION MEANS

BIPROPELLANT VALVE DESIGN LAYOUT
 A. WEIGHT
 B. ENVELOPE
 C. MAINTAINABILITY
 D. PURGEABILITY
 E. RELIABILITY
 F. MATERIALS
 G. OTHER CONSIDERATIONS

QUADVALVE LAYOUT
 A. ENGINE INTERFACES
 B. SYSTEM INTERFACES
 C. WEIGHT
 D. ENVELOPE

4-28

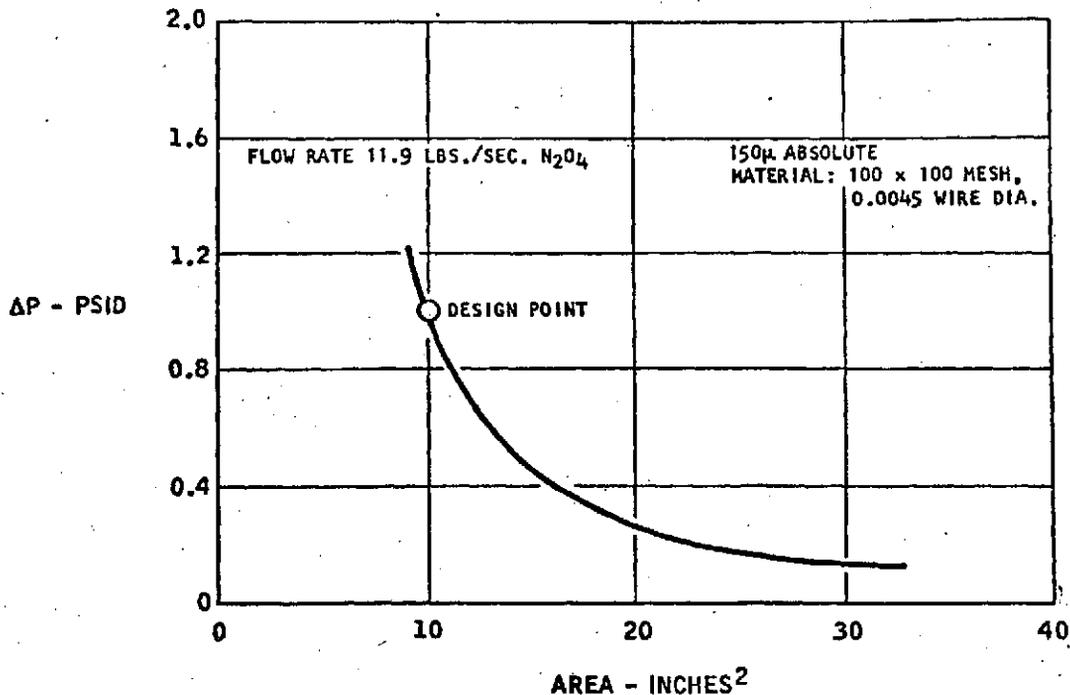
Figure 4-13

During the performance of these analyses tasks a bipropellant valve design layout was prepared to evaluate the ability to accommodate the size of valve components specified by the analysis such as bellows, springs, sealing closure configuration, reduction gears, etc., as well as to determine the weight and envelope of the bipropellant valve. In addition the valve design layout addresses such requirements as maintainability, purgeability, reliability, materials of construction, and others. The analysis and the valve design layout efforts are of course an iterative process to arrive at an optimum bipropellant valve design. Upon completion of the bipropellant design layouts of the quad valve were prepared. These layouts addressed such requirements as engine and system interfaces as well as the mounting of the four bipropellant valves to one another. Representative analyses and design considerations that were accomplished during this program are presented in the following paragraphs.

Pressure Drop

The quad redundant valve package was originally conceived as a package of four bipropellant valves with a single fuel inlet, a single oxidizer inlet, a single fuel outlet, a single oxidizer outlet, a single fuel inlet filter, and a single oxidizer inlet filter. Since the total pressure drop available was 5 psid at 11.9 pounds per second of nitrogen tetroxide this pressure drop had to be split up among the inlet filter, manifold inturns, and the valves. In determining the pressure drop budget the filter pressure drop was analyzed first.

Based on the teflon sealing closure designs employed in both the dual poppet and the ball valves it was determined that the filter requirement should be 150 micron absolute. The required filter screen area as a function of allowable pressure drop for a flow rate of 11.9 pounds per second of oxidizer was therefore determined and is presented in Figure 4-14. It was considered desirable to accommodate the required filter screen area



FILTER ΔP VS EFFECTIVE FILTER SCREEN AREA

Figure 4-14

in the form of a pleated disc in the propellant inlet line so as to minimize the weight and envelope required for any filter housing. Design studies of the inlet tubing concluded that a filter area of 10 square inches could be accommodated. This meant that the filter pressure drop would therefore be 1.0 psid. Consequently this filter size was designated as the design point. Analysis of the number of bends required to go from the single propellant inlet to the two inlets of the particular propellant in the quad redundant valve package concluded that a pressure drop of approximately .24 psid would have to be accommodated. Consequently out of the total pressure drop budget of 5 psid 3.76 psid remained for the valves. This meant a pressure drop of 1.88 psid for each individual valve at a flow rate of 5.95 pounds per second.

The ball valve was sized as a simple orifice with a discharge coefficient of (.9). The ball flow diameter (D_b) was computed from equation given below

$$D_b = \left(\frac{4 \dot{W}}{\pi C_d} \right)^{1/2} \left(2 g \rho \Delta P \right)^{1/4}$$

where:

- D_b = diameter of the ball orifice
- \dot{W} = propellant mass flow rate
- C_d = orifice discharge coefficient
- g = 32.2 ft/sec²
- ρ = propellant density
- ΔP = pressure drop across ball orifice

The diameter of both the fuel and oxidizer valves were sized the same which is for the largest required flow namely for that of the oxidizer valve. The value assumed for the discharge coefficient is based on the constant area minimum diameter hole being a short pipe with flow losses equivalent to a 10% pressure drop. This is the case for the valve being wide open. For intermediate open positions the discharge coefficient will be lower and the flow area will follow the curve shown in Figure 4-15. However, since the valve is a two position device the partial open position is not important. In the case of the poppet valve the pressure drop determined by the partial pressure drops for the contractions-expansion and turns as the fluid passes through the valve. However, the valve was sized to keep the velocity very low except through the minimum area section. Therefore, the majority of the pressure drop is across the poppet. The pressure drop in the poppet valve is computed from the following:

Drop for 90° turn

$$\Delta P_1 = K_{90} \rho v^2 / 2g$$

where $K_{90} = .3$

v = propellant velocity

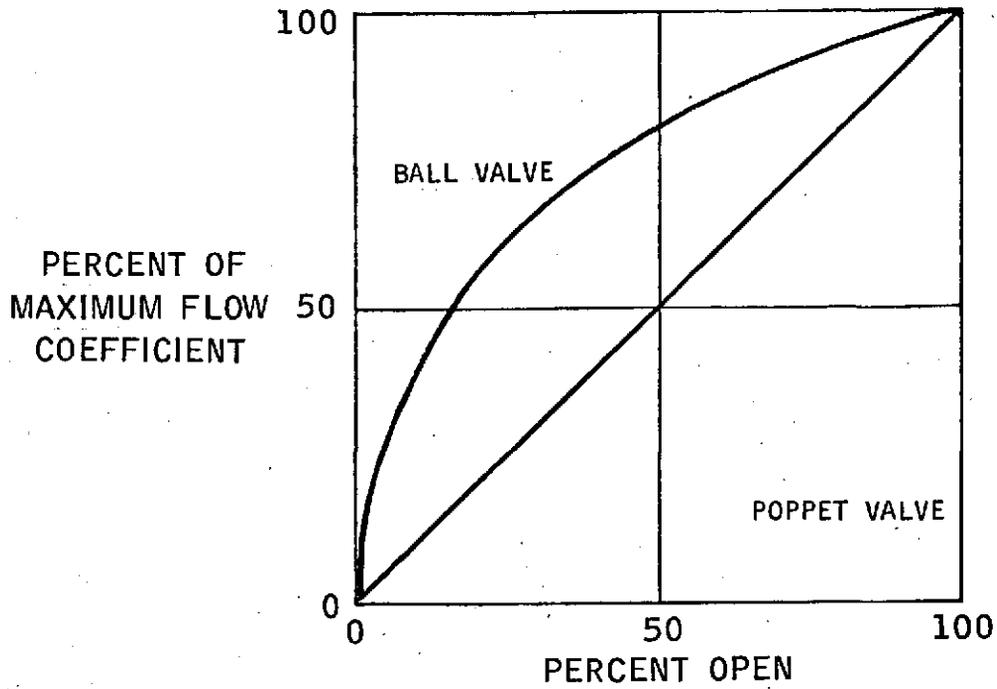
Drop across poppet land

$$\Delta P_2 = \frac{Q^2}{(C_d A)^2 2 g \rho}$$

where $C_d = .7$

Q = volumetric flow rate

A_v = flow area at land



EFFECTIVE FLOW AREA VS PERCENT VALVE OPEN

Drop in and out of valve

$$\Delta P_3 = K_0 \rho \frac{v^2}{2g}$$

where $K_0 = 1.0$

The total pressure drop which includes two 90° turns is

$$\Delta P = .875 \left[\frac{2 K_0}{A_0^2} + \frac{2 K_{90}}{A_{90}^2} + \frac{1}{(C_d A_v)^2} \right]$$

Leakage

The Marquardt Company has been engaged in a number of fluid system component technology programs requiring low leakage valves and pressure regulators and a result of these programs has developed a thorough understanding of sealing closure interface requirements. The leakage requirement for the OMS engine valve is a maximum of 10 scc per hour of helium over an inlet pressure range of 0 to 250 psia. Since the valve is required to operate over a relatively narrow operating temperature range it was concluded that this low leakage requirement could be readily met with the trapped teflon sealing closure design utilized by the Marquardt Company in a number of its valve designs. This design essentially consists of a 0.025 to 0.060 inch wide teflon sealing land which is backed up by a metal land and this configuration has been employed by Marquardt in flat, conical, and spherical sealing closure interfaces. Consequently the trapped teflon seal design is applicable to both the poppet valve and the ball valve.

The leakage model demonstrated by the Marquardt Company (reference 2) consists of summary laminar and molecular flow through a sealing closure interface and is represented by the following equation.

$$Q = Q_L + Q_M$$

$$Q = \frac{1.46 \times 10^3 D_S (P_1^2 - P_2^2) H^3}{\mu L T} + \frac{1.55 \times 10^8 D_S (P_1 - P_2) H^2}{L}$$

where

Q_S	Leakage Rate
D_S	Diameter of Seat
P_1	Upstream Pressure
P_2	Downstream Pressure
L	Seat Land Width
T	Gas Temperature
ψ	Coefficient
σ_S	Seating Stress
σ_Y	Yield Strength of Softer Material
μ	Viscosity
h	Peak to Valley Height
H	Effective Leak Path Height

$$\text{where } H = (h_1 + h_2) \left(1 - \frac{\sigma S \psi}{\sigma Y} \right)$$

As evidenced from this leakage model, the surface finish is the most important parameter in establishing low leakage through a sealing closure since it is also desirable to maintain relatively low seating stresses to minimize actuation force requirements and to enhance sealing closure life a peak to valley height (h) of 1.4×10^{-6} inches corresponding to a surface finish of 2 AA was determined.

Materials of construction

Material selection is a part of the design configuration process functioning primarily to screen and eliminate candidate materials with inadequate physical, chemical or metallurgical properties. The operating temperature and pressure requirements for the OME shutoff valve and actuation system are not particularly severe, and material selection was therefore influenced primarily by the long term exposure (10 years) and the compatibility with the operating fluids and the salty air environment.

The Marquardt Company has been engaged in the selection of materials for a number of space shuttle components. These include the earth storable bipropellant RCS engine and injector valve, as well as the helium pressure regulator for the OMS. Consequently, The Marquardt Company has accumulated a substantial data file on candidate materials and has also published data such as Reference (1) in this area. A summary chart presenting the most promising materials for the OME shutoff valve and actuation system is shown in Table 4- VIII.

The long term application of the OME shutoff valve places particular emphasis on the evaluation of materials for crevice corrosion, fatigue factors, stress corrosion sensitivity, and galvanic couples. It also makes the use of age controlled materials undesirable.

The OME shutoff valve must be compatible with nitrogentetroxide and monomethylhydrazine. Even more important to the selection of materials is their compatibility with the reaction products of these propellants, with one another, as well as with moist, salt air. Finally, the valve must also be compatible with anticipated cleaning and flushing fluids.

The partial reaction of the oxidizer and fuels tend to form amine nitrates. These are generally more corrosive than the propellants individually. Under most adverse conditions of low temperature and high vacuum, amine nitrate mixtures which tend to be shock sensitive may be formed. In examining potential engine configurations it was apparent that these products could be located anywhere downstream of the valve sealing closure and throughout the injector and thrust chamber.

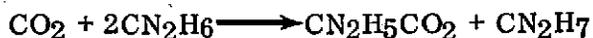
TABLE 4-VIII

CANDIDATE MATERIALS FOR SPACE SHUTTLE OME SHUTOFF VALVE

Item No.	Material	Principal Producers	Material Condition	Heat Treatment	Room Temp. Mechanical Prop.				Weldability	N ₂ O ₄ (Orig.) Compatibility	N ₂ O ₄ (3.2% H ₂ O) Compatibility	HNO ₃ Compatibility	H ₂ O (Mop) Compatibility	Fracn & Alcohol Compatibility	Trichloroethylene Compatibility	Amine Nitrates Compatibility	Amine Compatibility	Stress Corrosion Resistance	Crevice Corrosion Resistance	Gal. Corf. Potential EMF, Volts
					F _{TU}	F _{TY}	% Elong.	% Ha												
1	Inconel 718 Ni Base	Special Metals Huntington Alloy Cabot Corp.	Sol. Treat and Aged	1750°F/1 hr/RT to 1350°F/8 hr/PC to 1150°F/1150°F/8 hr/Air Cool to RT	180	150	10	35	Excellent	Compatible	Compatible	Good	Excellent	Excellent	Compatible	TBD	Compatible	Immune	Excellent	-0.15
2	Inconel 625 Ni Base	Huntington Alloy Cabot Corp.	Annealed	2100°F. Sol. Anneal	135	75	45	-	Excellent	TBD	TBD	Good	Excellent	Excellent	Compatible	TBD	TBD	Immune	Excellent	-0.15
3	6 Al-4V Ti Alloy	TMCA Reactive Metal Harvey Aluminium	Annealed	1400°F/1 hr/Cool to RT	130	120	10	-	Satisfactory	Compatible	Compatible	Good	Excellent	Excellent in alcohol. Incompatible to Frasn MF. Compatible in Frasn TB.	Incompatible	Compatible	Compatible	Excellent	Excellent	-0.15
4	K801 K96	Kenametal Corp.	Not Heat Treatable	Not Heat Treatable	50	Comp/St 690	Knoop Hardness 1050	-	Not Weldable Brazable	TBD	TBD	TBD	Excellent	Excellent	Compatible	TBD	TBD	Immune	Immune	+0.05
5	B4C	Max Labs	Not Heat Treatable	Not Heat Treatable	50	414	3000	-	Not Weldable Brazable	TBD	TBD	TBD	Excellent	Excellent	Compatible	TBD	TBD	Immune	Immune	+0.05
6	Cermet Au-Ni-Cr Braze	Handy & Harmon	Not Heat Treatable	Not Heat Treatable	-	-	-	-	Not Applicable	Compatible	Compatible	Excellent	Excellent	Excellent	Compatible	TBD	Compatible	Excellent	Excellent	+0.15
7	Micro Au-Ni Braze	Handy & Harmon	Not Heat Treatable	Not Heat Treatable	-	-	-	-	Not Applicable	Compatible	Excellent	Excellent	Excellent	Excellent	Compatible	TBD	Compatible	Excellent	Excellent	+0.15
8	AMS 4176 Ni Braze	Wall Colmoney	Not Heat Treatable	Not Heat Treatable	-	-	-	-	Not Applicable	Compatible	TBD	Good	Excellent	Excellent	Compatible	TBD	Compatible	Excellent	Excellent	-0.15
9	304L SS Fe Base	Allegheny-Ludlum Steel Carpenter Steel Universal-Cyclops	Annealed	1950°F Solution Anneal	85	30	50	60	Excellent	Compatible	Fair	Fair	Good	Excellent	Compatible	TBD	Compatible	Excellent	Fair	-0.20
10	347 SS Fe Base	Allegheny-Ludlum Steel Carpenter Steel Universal-Cyclops	Solution Treated	1950°F Solution Anneal	90	35	50	85	Excellent	Compatible	Good	Good	Good	Excellent	Compatible	TBD	Compatible	Excellent	Good	-0.20
11	444 Fe Base	Allegheny-Ludlum Steel Alnico-Temesal	Annealed	Anneal to Brinell 207 Maximum	85	35	33	-	Marginal	Compatible	Good	Good	Good	Excellent	Compatible	TBD	Compatible	-	Good	-0.40
12	TFE	Marquardt Processed	Extruded and Sintered		3 to 4.2		300			Compatible	Excellent	Excellent	Excellent	Excellent	Excellent	Excellent	Excellent			
13	AFE-124D	TRW Processed	Molded		2.5					Compatible	TBD	Excellent	Excellent	Alcohol Excellent	Frasn Incompatible	Excellent	TBD	Good		

4C-4

The general concern with the possible reaction of propellants with the moist, salt air did not exist during the Apollo Program because of the single mission requirement; however, for the space shuttle program this is indeed a very real requirement due to the multiple mission usage. One potential problem which was investigated by Aerospace Corporation is the formation of carbazic acid due to the reaction of the amine fuels and the carbon dioxide in the atmosphere. These reactions are in accordance with the following relationships.



The problems encountered by Aerospace Corporation with carbazic acids were the plugging of monopropellant engine catalyst beds and the deposition of salts on the sealing surfaces of the rocket engine injector valve. The salts in the valves in turn caused valve leakage. The shutoff valves used on the Apollo LEM ascent engine, LEM descent engine, and the SPS all utilized ball-type sealing closures wherein the ball would slide over the seal during the opening and closing motions. This type of design results in significant wear even in a clean environment. It is apparent that salt deposits located immediately downstream of the sealing closure interface during the space shuttle program will greatly accelerate the wear in these types of valves. Consequently, the elimination of sliding valve sealing closures and the substitution of lift-off type seals were considered necessary for the space shuttle shutoff valve application.

The reaction of nitrogentetroxide with the moist air results in the formation of nitric acid. Consequently, material compatibility with nitric acid was also ascertained. Finally, the fact that the air contains salt and water required the consideration of the effect of condensed water and salt on cold surfaces.

Preliminary selection of valve materials has concluded that those materials listed in Table 4-VIII are most effective. For structural components such as valve housings, the titanium alloy appears most attractive because of its low weight and its excellent galvanic compatibility with stainless steels and nickel base alloys. For higher strength components and other items such as bellows Inco 718 appears most promising. The 300 series stainless steels are attractive for lower strength applications requiring good weldability. Type 446 stainless steel is a good choice as a magnetic material for solenoid actuators. Gold or gold/nickel alloys appear most promising for brazing applications. Of the plastic and elastomeric materials only Teflon appears to be fully compatible with the propellants and their reaction products and also with the 10-year life requirements. AFE-124D was also considered as a very promising seal material; however, long term compatibility data for this material is not yet available.

Maintainability and Purgeability

During the preparation of the valve designs a significant effort was made to incorporate into the ball and poppet valve features that would not require maintenance during the long ten year service life of the space shuttle. This included the utilization

of such components as metallic flexure guidance, flat poppet/seat interface, and liftoff seals to eliminate scrubbing and wear during valve operation. Nevertheless it was recognized that during the mission life of the valves it was quite conceivable that environmental conditions resulting from mishaps which considerably exceed the design capability could be imposed; consequently it was desirable to have designed the valve concepts so as to permit access to critical component elements such as sealing closures, dynamic seals, moving parts, and electro mechanical or electronic sub elements. For this reason the valve concepts included a minimum number of flange joints featuring redundant static seals to gain the desired component accessibility.

Each bipropellant valve was considered an individual unit such that it could be readily removed from the quad valve package or operated independently of the other bipropellant valves to assess proper valve operation. Test ports were also included in the quad redundant valve package to permit leak checking of the individual bipropellant valves.

The long life and 100 mission requirements of the space shuttle make decontamination and handling considerations an important aspect of the design. The present baseline decontamination procedure established by Rockwell International at the end of each mission consists of a helium gas purge during re-entry, followed by a hot GN2 purge on the ground, and subsequent storage of the hardware away from the launch pad, inside a building. However, considerable debate has taken place regarding whether it is really necessary to remove the propellants from the OME after each mission.

If the purging procedures are not entirely effective, such that small residuals of propellant remain in cavities and crevices of the shutoff valve, the potential of corrosion resulting from the combinations of these residues and moist air is much greater than if the valves had been simply left full of propellant. On the other hand, the fact that the OME shutoff valve will be relatively free of propellants permits numerous functional actuations of the shutoff valve which would present a problem if the propellant valves were left full of propellant. In this latter case, a functional checkout would consist of an engine burn.

To enhance the capability of purging the propellant valves components that are difficult to clean or decontaminate such as welded bellows and other parts featuring small clearances and crevices have been avoided. The flow path through the valves has been designed such as to present as clean a flow path as possible and to eliminate dead end passages and crevices whenever feasible. Since some relatively dead volumes could not be eliminated both the poppet and ball valve designs feature purge passages. These purge passages take advantage of the dynamic head of the purge gas by locating the inlet to the purge passages such that it is subjected to the total pressure and then routing the purge gas back into the relatively dead cavities such that a continuous circulation through these cavities takes place when the valve is in the open position. This technique makes available a purge differential pressure of 2 psi when gaseous nitrogen is flowed through the valve at 0.5 pounds per second at a pressure of 100 psia.

Lubrication Approach

The 10 year life and zero maintenance requirement for the space shuttle OME valve and actuation system made it desirable to incorporate into the valve design features which would not require lubrication or which would employ some form of permanent lubrication. In the valve section of the dual poppet valve it was possible to eliminate the need for any lubrication by incorporating metallic flexure guidance. In the case of the ball valve section this technique was not possible since the ball valve employs rotary motion and since metallic flexures capable of rotating through 90° are not available. A review of potential lubricants for the propellant cavity resulted in the conclusion that some form of teflon would be most suitable since it is fully compatible with the propellants and since long term exposure had been demonstrated. One form of teflon lubrication that was used in the Apollo SPS engine valve are armalon bushings which feature a fiberglass base impregnated with teflon. Other techniques to obtain a teflon impregnated surface include canidizing for titanium and such processes as tuftram, nituff, and lemcoloy for aluminum. The canidizing process has been previously employed on a moon probe and in the Viking program. The teflon impregnated aluminum was used in the LEM ascent engine.

A comparison of potential lubrication approaches for the actuator housing which is isolated from the propellants included the use of silicon oil, solid lubricants, and the teflon impregnation discussed in the preceding paragraph. Discussions with the suppliers of the silicon oil and the solid lubricants disclosed that neither had been used previously for a ten year application although both appeared promising as long as they were sealed in the actuator housing. Since the teflon impregnation did not require a sealed actuator housing and appeared to be the best technique for preventing uneven distribution of a lubricant it was selected for a subsequent incorporation into the load carrying surfaces in the actuator. These included thrust bearings, radial bearings, gears and cams.

Actuator

In selecting an electrical motor actuator for use in driving the bipropellant valve, consideration was given to two types of devices. These are conventional DC servomotor and DC torquer or torque motor (brushless). The conventional, relatively high RPM DC servomotor has the obvious advantage of high torque with low weight and power consumption. However, the disadvantage with this device is the inefficient power transfer from the motor to the load. Therefore, it is difficult to drive the motor backward through the gear box as is required to achieve fail safe capability. The DC torque motor with continuous rotation was selected as the best compromise in weight, power consumption and need for a small or no gear box.

After selection of a DC torque motor it was necessary to evaluate the advantages and disadvantages of brush vs. brushless types. The principal advantage of the brush type DC torque motor over the brushless torque motor is that it requires no electronic driving circuitry and is slightly lower in power consumption and weight (about 10%).

The evaluation of brushless motors was limited to those using Hall generators to implement solid state commutation since these have a history of successful usage in space applications.

The primary advantages of the brushless motor are listed below.

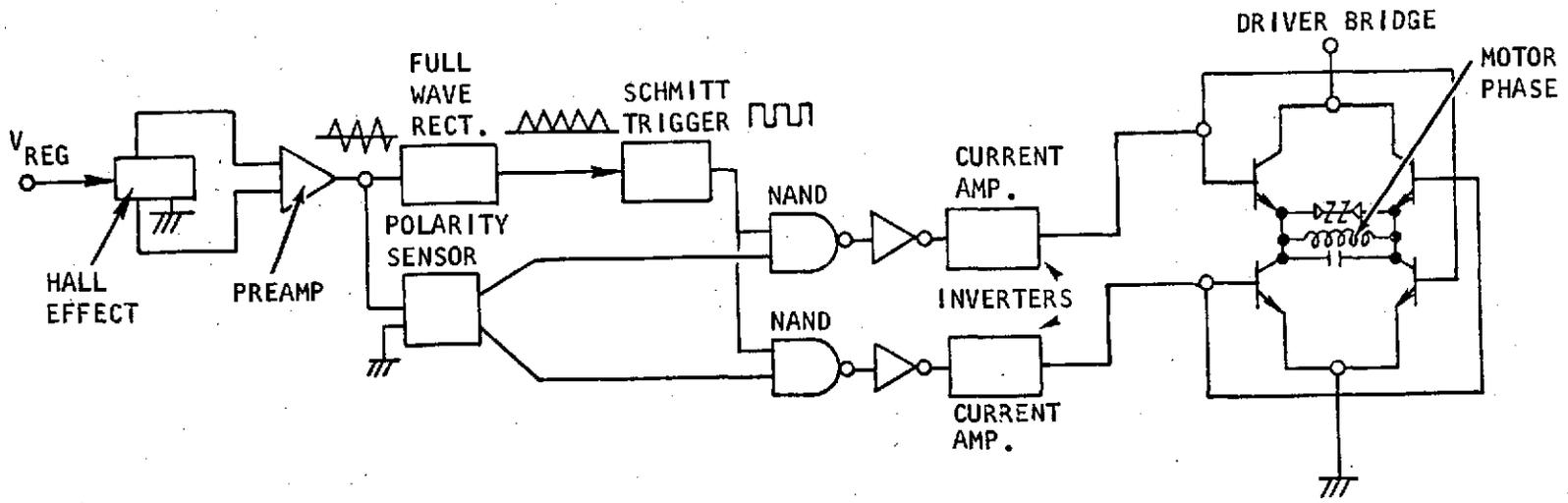
- 1) No brushes eliminates friction which results in two desirable features especially in this application - infinite life and no contamination created by brushes.
- 2) Explosion proof with no arcing and no electrical interference and minimal acoustic noise.
- 3) Although it is not extremely important in this application, there are essentially no torque dead spots as are found in brush commutated motors.

The Hall effect brushless DC torque motor has been successfully used in several different space applications including the following:

- 1) Apollo - antenna steering motor
- 2) Communication satellites - Despin antenna (60 rpm for 7 years)
- 3) Ranger and Mariner - jet vane actuator
- 4) Lunar walk (motor used in back pack)

In addition, the same type motor is presently being used in experiments to develop an implantable heart pump machine.

The primary disadvantage of the Hall effect brushless motor is the requirement for a small electronic circuit. This circuit in effect replaces the brushes in a brush type motor and performs the commutation of the magnetic field. In the case of the two phase motor that was recommended by the supplier for this application the electronic driver receives signals from two Hall effect devices located in the motor to determine the location of the magnetic field and based on these signals alternately drives the two windings (phases) of the motor. The most efficient way of driving the brushless DC torque motor as far as the motor is concerned is by utilizing a driver with a sine/cosine wave output. This type of output results in the highest torque and minimum torque ripple. However a driver capable of generating sine/cosine outputs consumes a substantial amount of power itself and is larger in weight and envelope than a digital output driver. Consequently when the combination of motor and driver is considered a digital driver results in a lower overall power requirements particularly if the digital driver is designed to provide a 60% duty cycle which most closely simulates the sine/cosine wave. The digital driver is also more reliable and less costly than the sine/cosine wave driver. A typical digital torque motor phase control logic is presented in Figure 4-16. This figure also shows the wave forms as they exist in certain portions of the logic. Further insight into the driver logic may be gained by examining Figure 4-17.

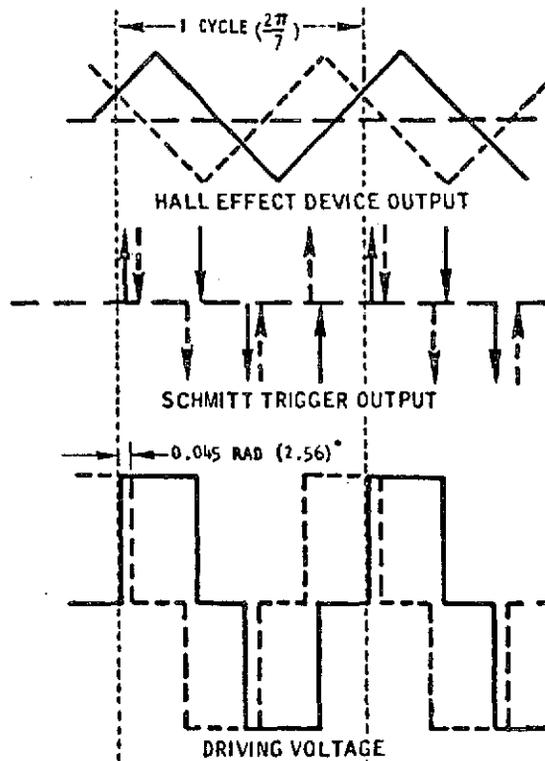


• NOTE: TWO OF THESE CIRCUITS ARE REQUIRED TO DRIVE ONE TORQUE MOTOR

TYPICAL TORQUE MOTOR PHASE CONTROL LOGIC

4-39

Figure 4-16

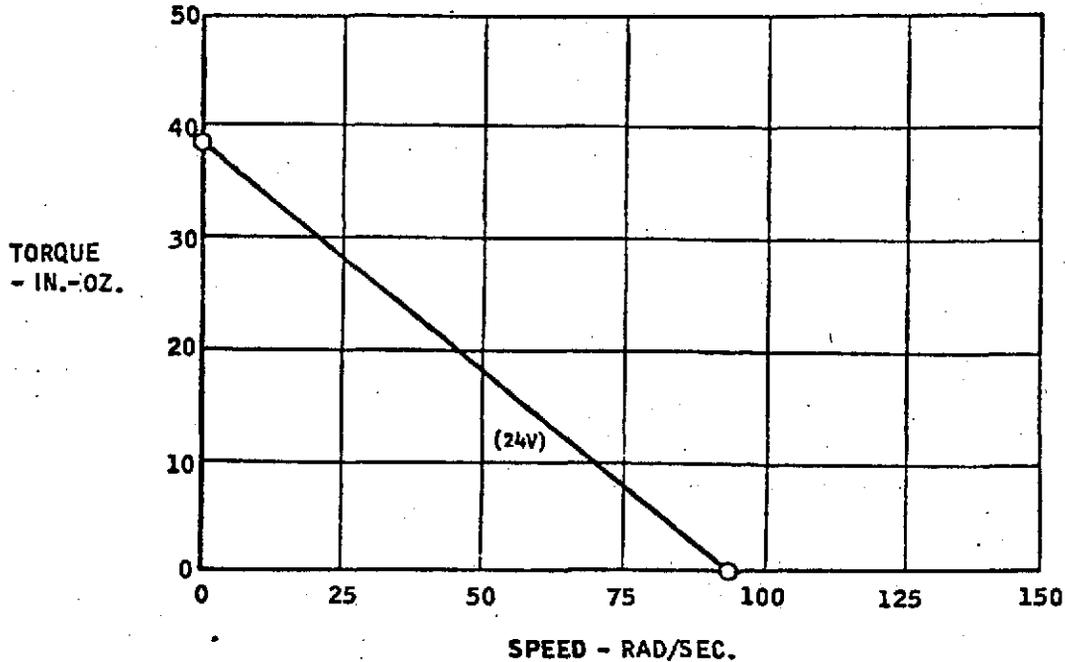


TORQUE MOTOR DIGITAL DRIVING SIGNALS

Figure 4-17

This figure shows the events that happen during one cycle of operation in the control logic. The solid lines in this figure represent the signals to one of the motor windings and the dashed lines represent the signals to the other motor winding. A 60% duty cycle is shown. The sawtooth wave at the top of the figure roughly simulates the output of the Hall effect devices as the magnetic field changes due to the rotation of the motor rotor. The Hall effect device output is sensed by a Schmitt trigger which in turn is used to start and stop the driving signal to each winding. By adjusting the Schmitt trigger level the square wave output to the windings may be adjusted to any desired pulse width. The windings are driven alternately in both directions by means of a polarity sensor in the circuit such that the output signals shown at the bottom of Figure 4-17 result.

Typical torque motor torque versus speed characteristics are shown in Figure 4-18. As shown here the motors feature linear torque-speed characteristics. Maximum torque is available at 0 speed and is called stall torque. As the maximum motor speed is reached the available torque approaches 0. Typically a motor operating point at approximately 50% of stall torque is chosen. During the performance of the tradeoff study several brushless DC torque motor suppliers were contacted to obtain parametric motor performance data. These data are plotted in Figures 4-19 and 4-20. It should be pointed out that the data plotted in Figure 4-19 is for a number of different torque motors at the operating condition of approximately 50% of stall torque. Since the torque/speed characteristics of these

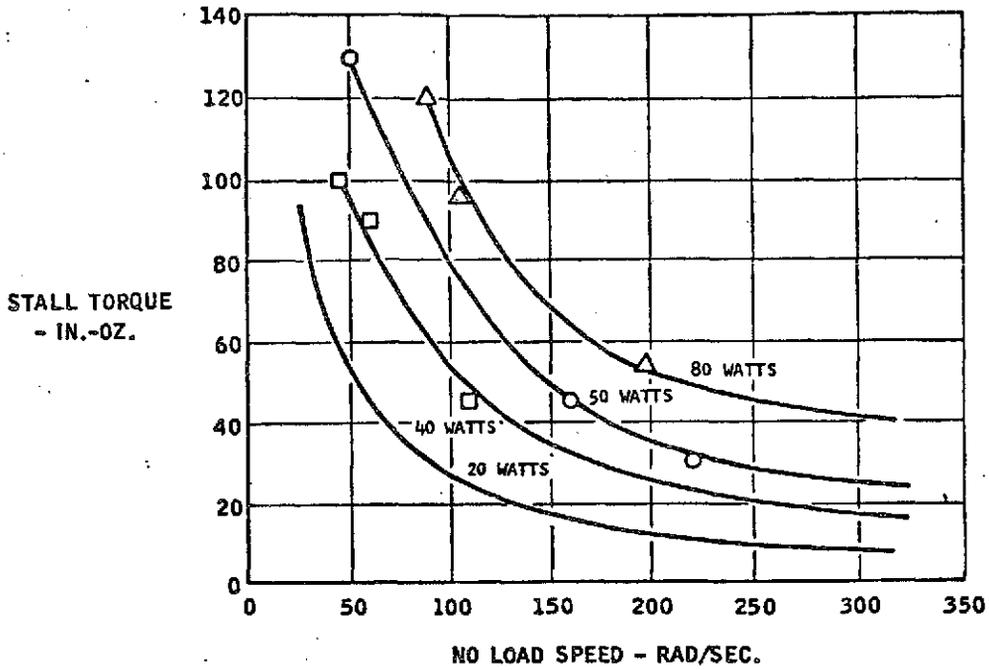


TORQUE/SPEED AT 40 WATTS

Figure 4-18

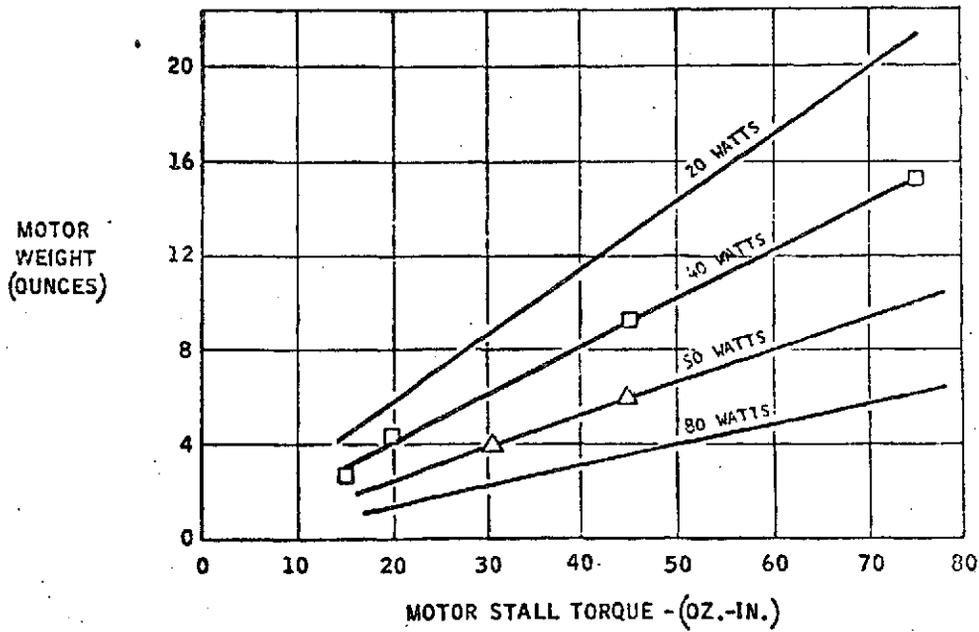
different motors do not feature the same slope as that shown for one typical motor in Figure 4-18 the resultant tradeoff curves are not linear. The data presented in Figures 19 and 20 was then used in optimizing the weight of the valve actuators, in selecting the best torque motor and gear ratio combination to achieve acceptable response characteristics and to minimize weight to limiting factors had to be considered. Since the valve had to be designed to fail safe it was necessary to maintain electrical power to the actuator as long as the valve was open and once the electrical signal was terminated to depend upon a spring to return the valve to the closed position. Since the closing spring had to be located on the valve shaft to achieve a reasonably sized spring it was necessary to back drive the torque motor through the actuator gears in the closing direction. Based on Marquardt's previous experience it was therefore concluded that the gear ratio should be limited to a maximum of 50 to prevent jamming up of the gear train during the closing motion. Also since the electrical power was required while the valve was open it was necessary to determine the maximum amount of energy that could be dissipated without overheating the torque motor or the Hall effect devices located at the working gap of the torque motor.

The limiting temperature of the Hall effect devices is 210°F. It was assumed that the maximum ambient temperature could reach 140°F. and that the heat generated in the torque motor would have to be transferred primarily by conduction to the body of the valve and to the propellant lines. Consequently a maximum allowable temperature difference of 70° between the torque motor and the valve body was established. Upon examining a number of the valve designs that were being compared during the tradeoff and determining the heat conduction path available from the motor to the valve housing it was concluded that the torque motor power consumption should be limited to 40 watts. In subsequent actuator



DC TORQUE MOTOR CHARACTERISTICS
STALL TORQUE VS NO LOAD SPEED

Figure 4-19



DC TORQUE MOTOR CHARACTERISTICS
MOTOR WEIGHT VS STALL TORQUE

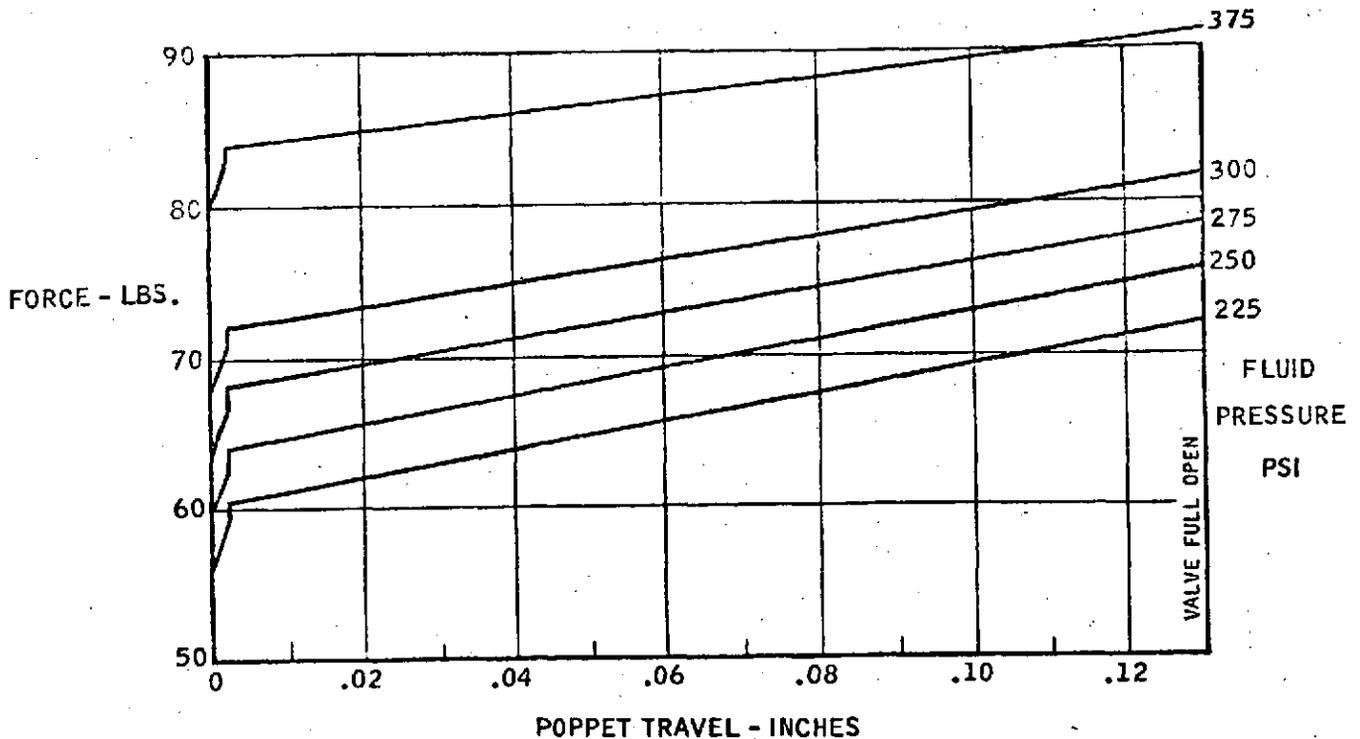
Figure 4-20

sizing to achieve the required response characteristics it was determined that the 40 watt power level was sufficient to meet response requirements.

Poppet Valve Sizing

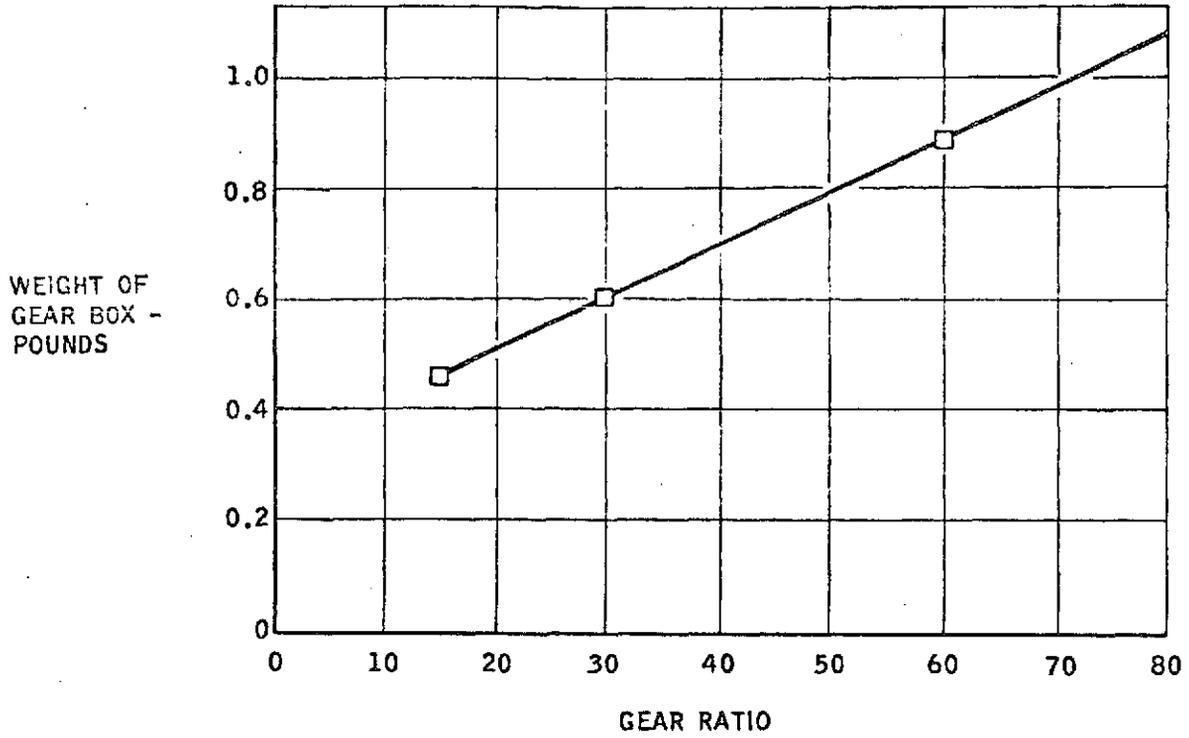
The dual poppet valve that was considered during the final tradeoff was previously described in Section 4.2.1 and was shown in Figure 4-8. A detailed analysis of this valve was made to optimize its weight. This analysis included sizing of the sealing closure interface, axial guidance flexures, bellows, and flow passages through the valve body. Once the size of the valve mechanism was determined the actuator was optimized with respect to weight by trading off gear ratios and torque motor sizes. Details of the dual poppet analysis are presented in Appendix A. Some of the highlights will be briefly presented in this section.

To permit the optimization of the valve actuator the force versus stroke characteristics of the poppet valve had to be determined. These characteristics are presented in Figure 4-21 as a function of various inlet pressures. As shown in this figure the actuation force requirement varied from 60 lbs. in the closed position to 75 lbs. in the full open position at a stroke of .13 inches for an operating pressure of 250 psia. Consequently an actuator had to be designed which provided two axially moving output shafts each of which was capable of providing a minimum of 75 lbs. force. Since a 100% force margin was used in this valve design the required actuator output was actually 150 lbs. per shaft. Parametric data was then prepared to determine the gear box weight as a function gear ratio (Figure 4-22) and the motor weight as a function of gear ratio (Figure 4-23) in order to optimize the actuation system weight. Figure 4-24 presents the conclusions of this tradeoff by showing



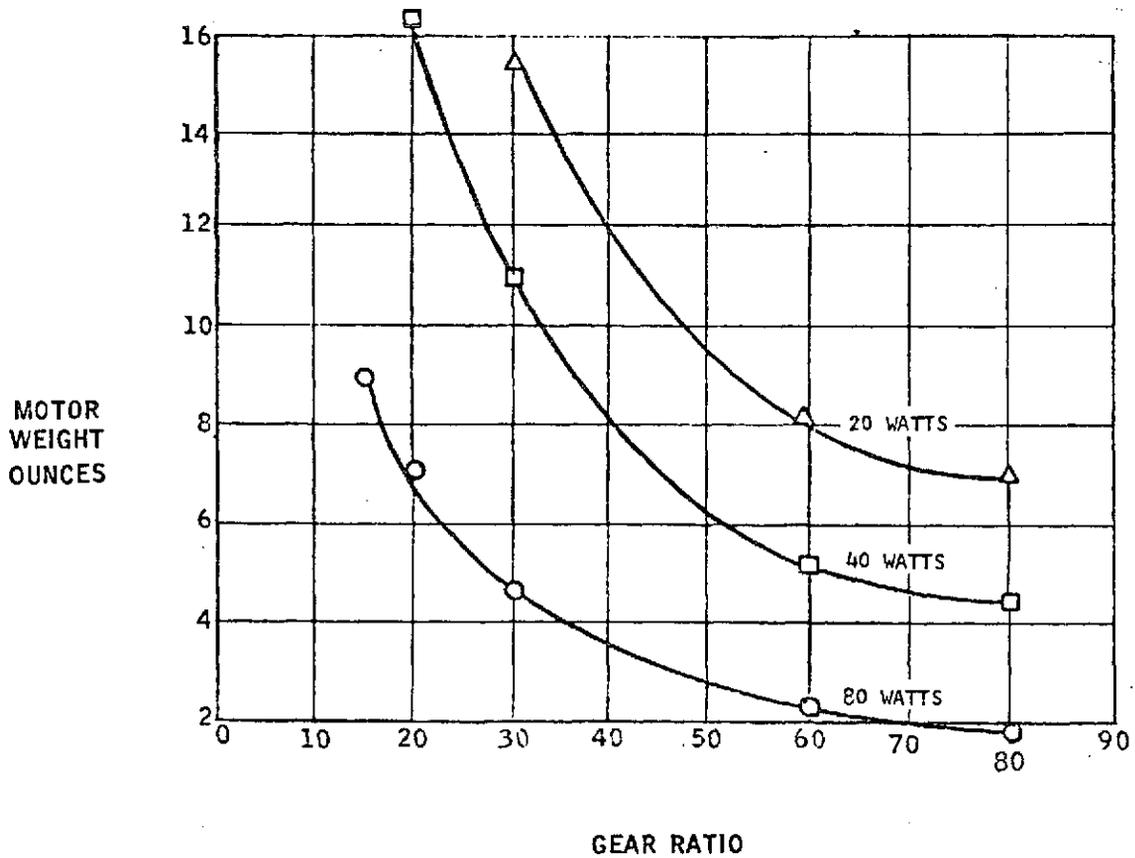
BALANCED POPPET VALVE
MAXIMUM FORCE REQUIRED VS STROKE

Figure 4-21



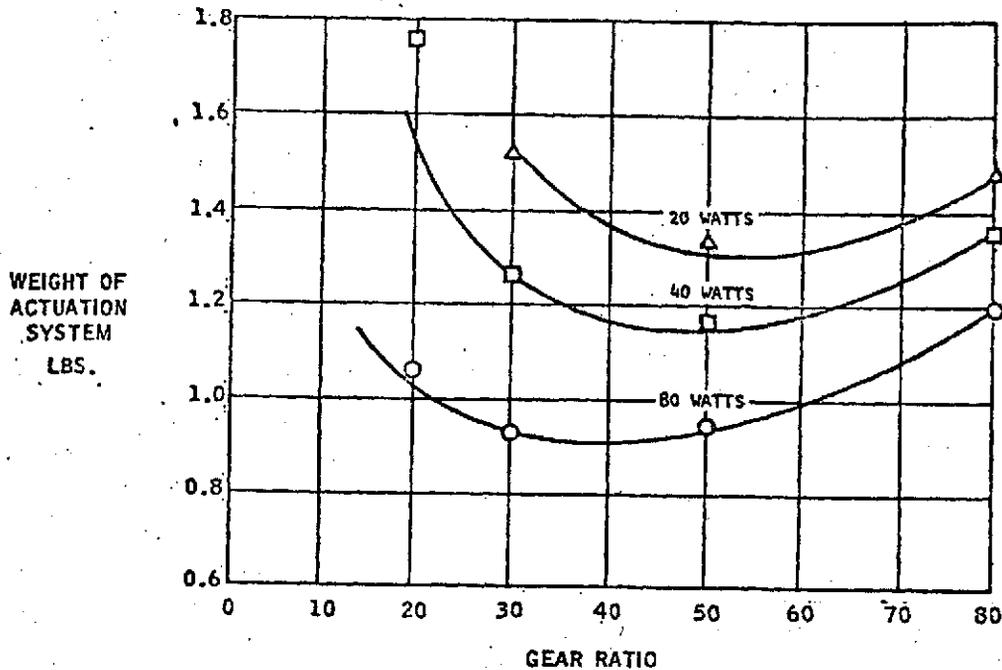
WEIGHT OF GEAR BOX VS GEAR RATIO

Figure 4-22



MOTOR WEIGHT VS GEAR RATIO

Figure 4-23



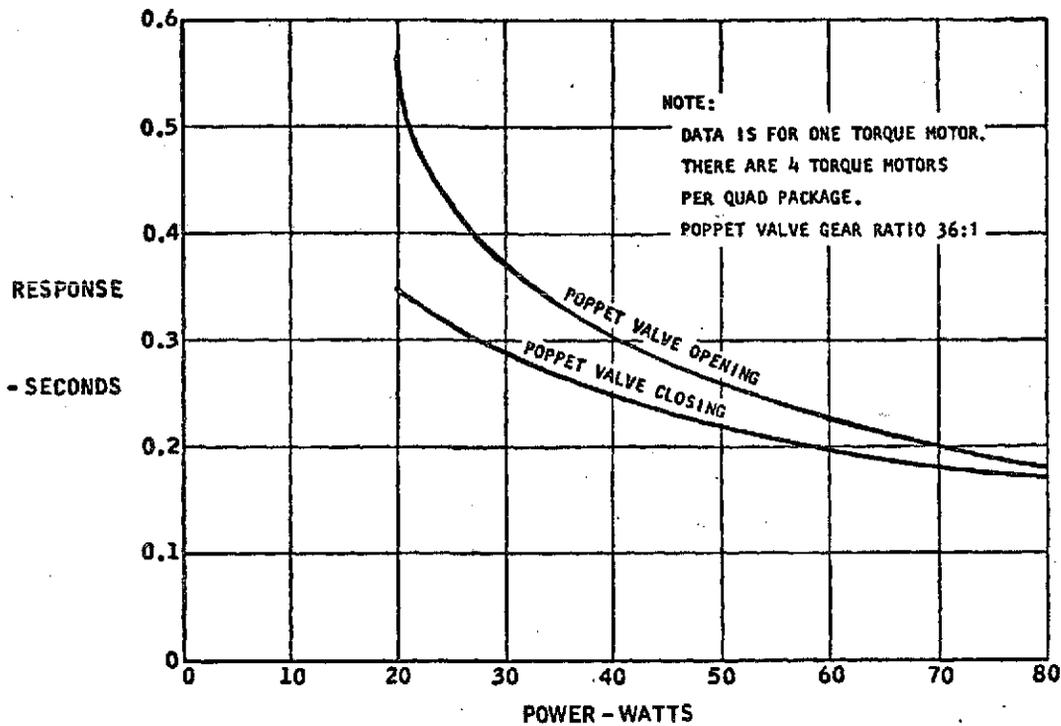
ACTUATION SYSTEM WEIGHT VS GEAR RATIO

Figure 4-24

the actuation system weight as a function of gear ratio at various torque motor power levels. Since it was desirable to limit the torque motor power level to 40 watts and to utilize gear ratios of less than 50 as discussed previously a design configuration featuring a 40 watt motor at a gear ratio of 36 and a weight of 1.2 lbs. was chosen.

The opening response of the valve is primarily a function of the velocity of the torque motor over the required stroke. This average velocity of the motor is computed from the torque/speed characteristics (such as Figure 4-18) assuming an average load torque (obtained from the average force shown in Figure 4-21). The total opening speed was then computed as the sum of the motor torque rise time (approximately 30 milliseconds) and the average velocity. The predicted poppet valve opening response is shown in Figure 4-25 as a function of torque motor operating power. The closing response is also shown in this figure. The closing response is a function of the net closing torque (resulting from the precompression forces of axial guidance flexures and the bellows in the valve mechanism) and the inertia of the complete valve. As evident from Figure 4-25 both opening and closing responses are quite acceptable at the 40 watt power level.

Upon completion of the valve sizing and the preparation of the valve design layout the final predicted valve and actuation system weight was determined. A breakdown of this weight is presented in table 4-IX. As evident from this table a total weight of 4.20 pounds per bipropellant valve was projected.



VALVE OPENING RESPONSE VS TORQUE MOTOR INPUT POWER

Figure 4-25

TABLE 4-IX DUAL POPPET VALVE AND ACTUATION SYSTEM WEIGHT BREAKDOWN

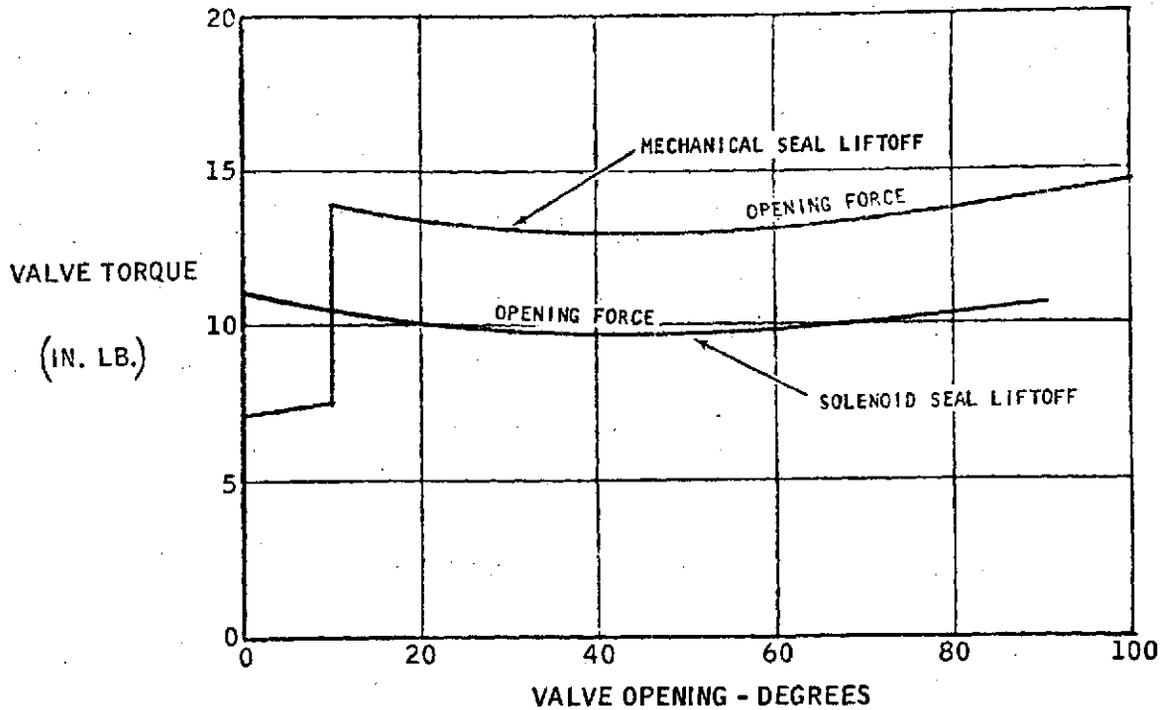
CONDITIONS:

PRESSURE DROP = 1.88 PSI PER BIROPELLANT VALVE
 FLOW RATE = 115 IN³ /SEC PER BIROPELLANT VALVE
 GEAR RATIO = 30:1
 POWER 40 WATTS

● POPPET (0.28) X 2	0.56
● VALVE HOUSING AND MANIFOLD (1.07) X 2	2.14
● ACTUATOR	0.57
● GEAR TRAIN AND LINKAGE	0.28
● ACTUATOR AND GEAR TRAIN HOUSING	0.40
● ELECTRONICS	0.25
TOTAL	4.20 LBS.

Ball Valve Sizing

The ball valve sealing closure was sized in accordance with the discussion presented previously under the heading of "Pressure Drop" and "Leakage". The torque required to operate the ball valve was then determined and is presented in Figure 4-26. Since the

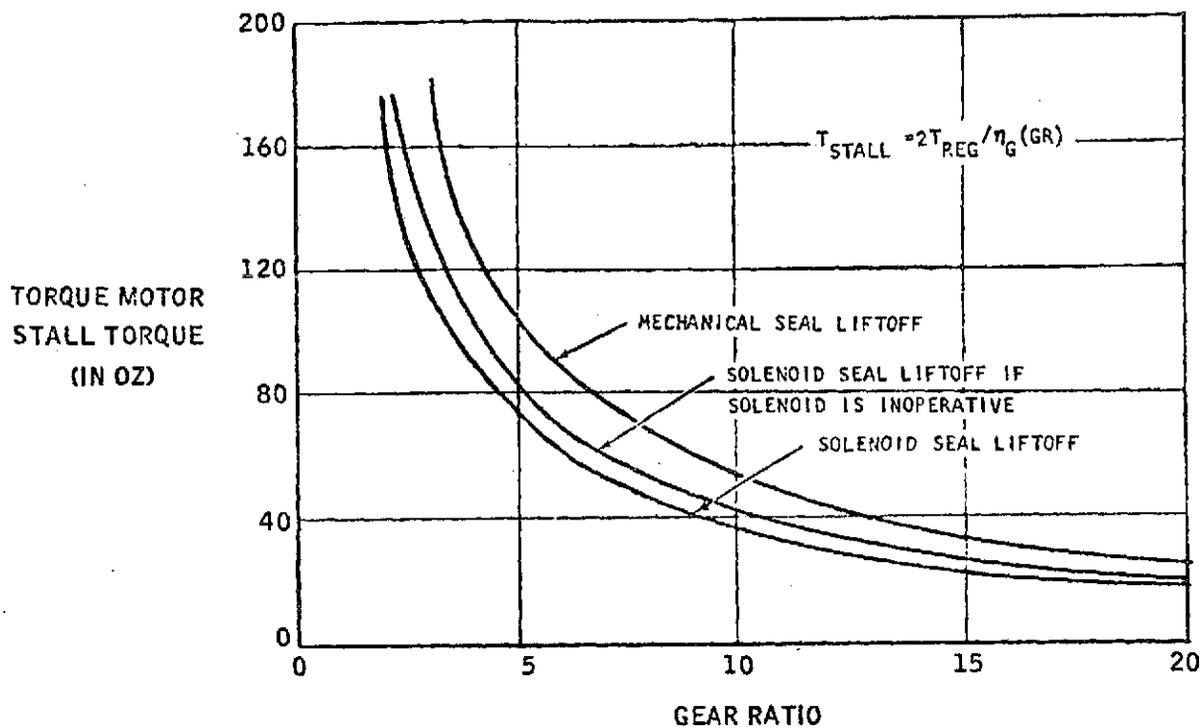


BALL VALVE MAXIMUM FORCE REQUIRED VS STROKE

Figure 4-26

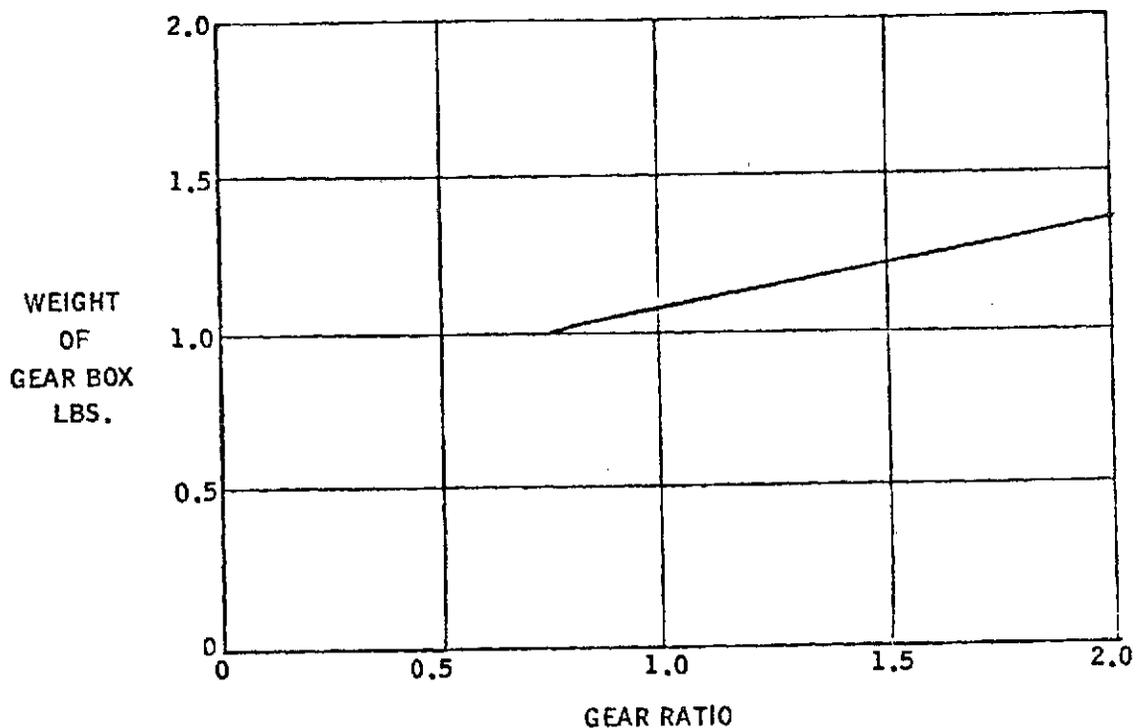
tradeoff study included both a solenoid retracted seal configuration and mechanically retracted seal configurations somewhat different operating torques were required for the two types of valves. As shown in Figure 4-26 the torque required for the mechanical seal liftoff is somewhat lower initially since the actuator output shaft is required to lift off the seal only during the first ten degrees. Subsequently the required torque is greater than that of the solenoid seal liftoff configuration since the friction of the mechanical lift off cam is not inherent in the solenoid seal liftoff configuration.

Based on the required ball valve operating torque a tradeoff was made to determine which gear ratio in combination with the torque motor would result in the lowest weight actuation system. Figure 4-27 shows the required torque motor stall torque as a function of gear ratio. Figure 4-28 presents the weight of the gear box as a function of gear ratio, Figure 4-29 shows the resultant torque motor weight as a function of gear ratio for a thirty watt torque motor, and Figure 4-30 presents the conclusion of this tradeoff



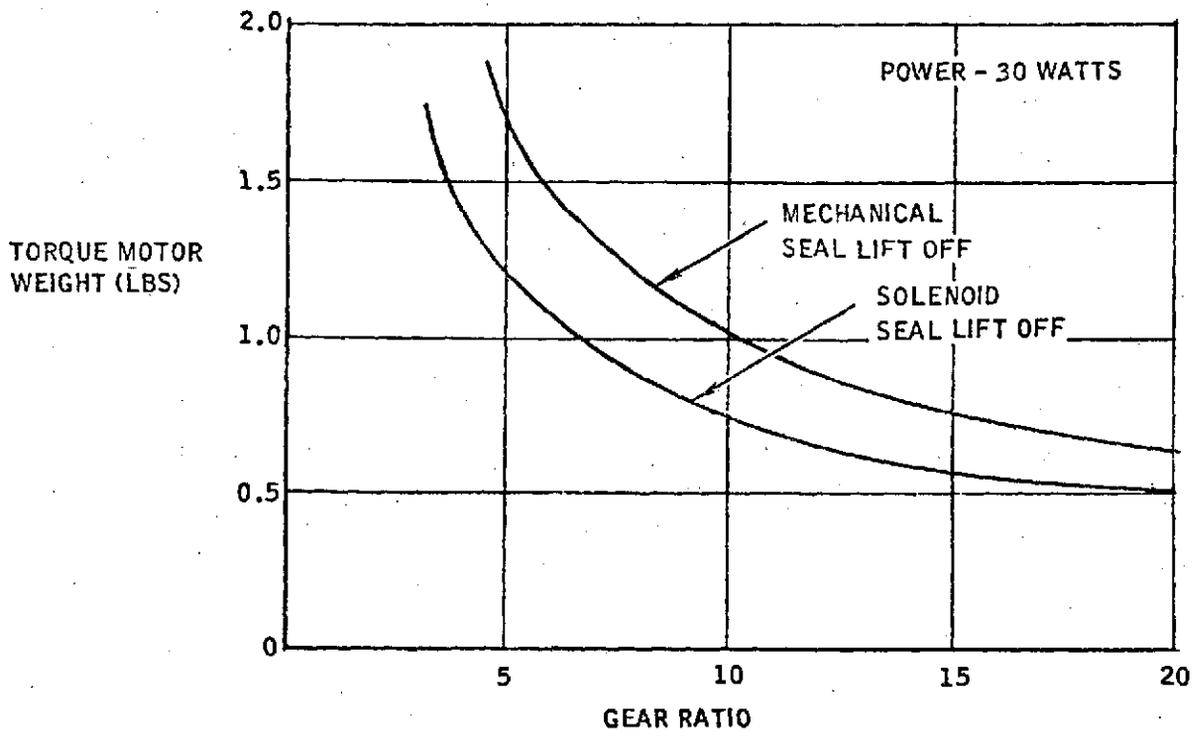
BALL VALVE TORQUE MOTOR STALL TORQUE REQUIRED VS GEAR RATIO

Figure 4-27



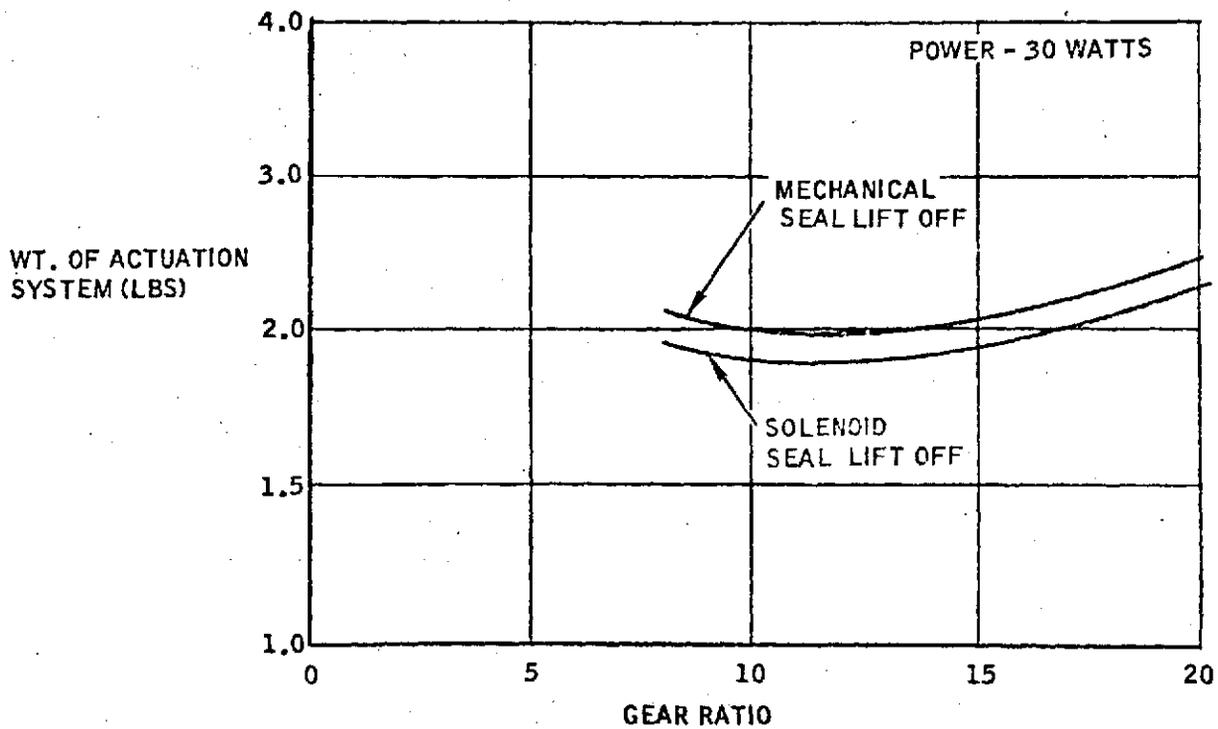
WEIGHT OF BALL VALVE GEAR BOX VS GEAR RATIO

Figure 4-28



BALL VALVE TORQUE MOTOR WEIGHT VS GEAR RATIO

Figure 4-29



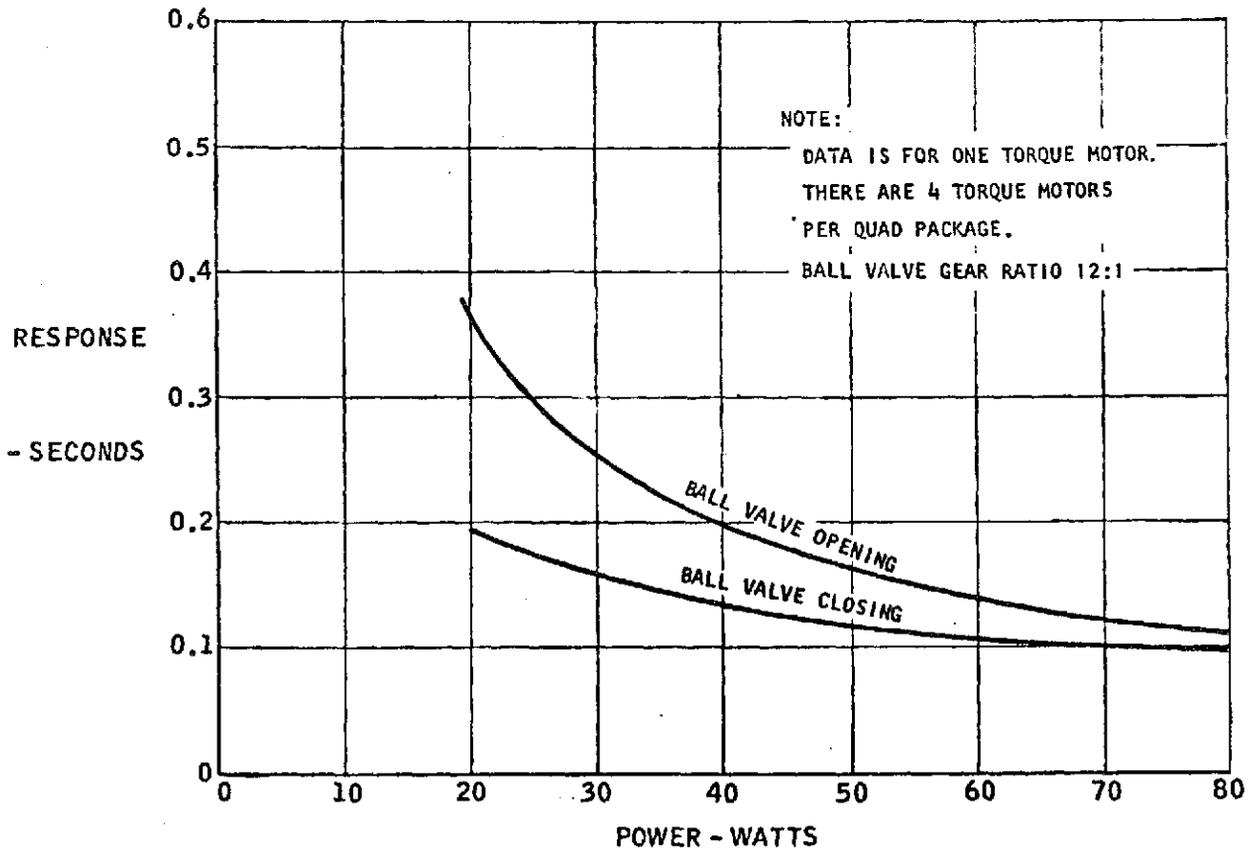
BALL VALVE WEIGHT OF ACTUATION SYSTEM VS GEAR RATIO

Figure 4-30

which is the actuation system weight as a function of gear ratio. As evident from this figure the lowest weight actuation system is achieved at a gear ratio of approximately 12 which was therefore incorporated into the valve design.

The design criteria for the solenoid which was required to operate the solenoid retracted seal configuration included a force requirements of 6 pounds, an operating stroke of 0.005", and a coil resistance of 56 ohms. The solenoid was designed to utilize type 446 stainless steel in the magnetic circuit and 300 series stainless steel in non-magnetic portions of the solenoid. The solenoid was sized to operate at a magnetic flux density of 8 kilogauss at nominal operating voltage and temperature.

Response characteristics for the ball valve were determined in a manner similar to that for the poppet valve and the resultant characteristics are presented in Figure 4-31. According to this figure the predicted response for the ball valve was 250 milliseconds opening and 160 milliseconds closing with a 30 watt motor.



VALVE OPENING RESPONSE VS TORQUE MOTOR INPUT POWER

Figure 4-31

The final layout of the bipropellant ball valve reflected the sizing presented in the preceding paragraphs and was utilized to determine the resultant valve weight. Table 4-X presents the ball valve weight breakdown. Accordingly the solenoid seal liftoff configuration is 0.15 lbs. heavier than the mechanical liftoff configuration due to the inherently heavier solenoid actuator.

4.2.3 Final Tradeoff Conclusions

A summary of the major performance characteristics of the four valve configurations compared during the final tradeoff analysis is presented in Table 4-XI. This table shows that the electrical power requirements of the ball valves featuring mechanical seal retraction are somewhat lower than those of either the dual poppet valve or the ball valve featuring solenoid seal retraction. In fact the additional electrical power required by the solenoid makes that valve configuration least desirable from an electrical power consumption point of view. However, all of the electrical power requirements are considered reasonable and therefore did not have a major impact on the final selection. With respect to weight the dual poppet valve was determined to be significantly lighter than the other configurations. Weight differences between the three ball valve versions were almost negligible.

TABLE 4-X
BALL VALVE WEIGHT AND ACTUATION SYSTEM BREAKDOWN

CONDITIONS:

PRESSURE DROP = 1.88 PSI PER BIROPELLANT VALVE
 FLOWRATE = 115 IN.³/SEC. PER BIROPELLANT VALVE
 GEAR RATIO = 12:1
 ACTUATOR POWER = 30 WATTS

	SOLENOID SEAL LIFT OFF	MECHANICAL LIFT OFF
● BALL	0.35	0.35
● SOLENOID	1.70	-----
● LIFT OFF (MECHANICAL)	-----	1.31
● SHAFT	0.22	0.22
● HOUSING (VALVE)	2.13	2.38
● ACTUATOR	0.57	0.57
● GEAR TRAIN	0.39	0.39
● HOUSING (ACTUATOR AND GEARS)	0.18	0.18
● ELECTRONICS	0.25	0.25
	<hr/>	<hr/>
TOTAL	5.80	5.65

TABLE 4-XI
FINAL TRADE OFF DATA COMPARISON
QUADVALVE PACKAGE

VALVE TYPE	POWER (WATTS)	WEIGHT (LBS.)	RESPONSE (MS)	
			OPEN	CLOSE
• DUAL POPPET (BALANCED)	160	17.0	300	250
• BALL (EXTERNAL SEAT RETRACTION)	120	23.2	250	160
• BALL (INTERNAL SEAT RETRACTION)	120	22.8	250	160
• BALL (SOLENOID SEAT RETRACTION)	232	23.4	250	160

ALL VALVES FEATURE BRUSHLESS DC TORQUE MOTORS

As far as the response characteristics of the valves are concerned the ball valves featured nearly identical response characteristics which were slightly faster than those of the dual poppet valve. Again all of the response characteristics were considered very reasonable.

A final comparison was then made by assigning a certain number of points to the various valve performance characteristics. A summary of this comparison is presented in Table 4-XII. The various performance characteristics rated are listed in the column on the left and these have been assigned certain weighting factors which are presented in the second column. Thus in arriving at a point total for each parameter the weighting factor was multiplied by the number of points assigned for that factor. A maximum number of 10 points were permitted for each of the parameters. According to the weighting factors such parameters as weight, maintainability, reliability, and contamination sensitivity were considered most important and were assigned a weight factor of 10. Electromagnetic interference characteristics of the particular valve were considered least important and were therefore only assigned a weighting factor of 5.

An example of the method used in arriving at the specific number of points listed under each of the valve headings is as follows. In the case of power consumption the ball valves featuring the mechanical seal liftoff were rated best and were assigned the maximum point total of 10 points. Since power consumption was weighted with a factor of 8 the number of points for this characteristic to each of the two valves was 8×10 or 80. Since the power consumption of the ball valve featuring a solenoid seal liftoff was highest it was assigned only 8 points for this characteristic. Again multiplying the weighting factor of 8 times the allocated number of points (8) resulted in a point total of 64 for this valve. Similarly all the other rating point totals were computed.

In accordance with Table 4-XII the comparison then showed the dual poppet valve to have the highest point total at 1054 with the ball with a solenoid seal liftoff in second place with a point total of 1023. Consequently the dual poppet valve was chosen for detail design and further development. However as a result of contractual changes initiated by NASA technical personnel the second place ball valve was also subsequently detail designed.

4.3. VALVE PRESSURE DROP TRADEOFF

Upon completion of the valve tradeoff study presented in the preceding sections and as a result of other tradeoff studies being performed by NASA and Rockwell International it became apparent that it might be more desirable to design the OMS engine shutoff valve and actuation system for a pressure drop of approximately $1/4$ of the originally specified value. The reason for this interest was that a pressure drop reduction through the valve of 3.75 psid would permit a pressure rating reduction of the propellant tank in the OMS by the same amount and the resultant tank weight savings less the resultant valve weight increase would result in an overall space shuttle program savings. OMS tradeoff considerations indicated that lowering of the valve pressure drop by one psi would result in a system weight savings of 6 pounds. Thus it was apparent if the pressure drop

TABLE 4-XII

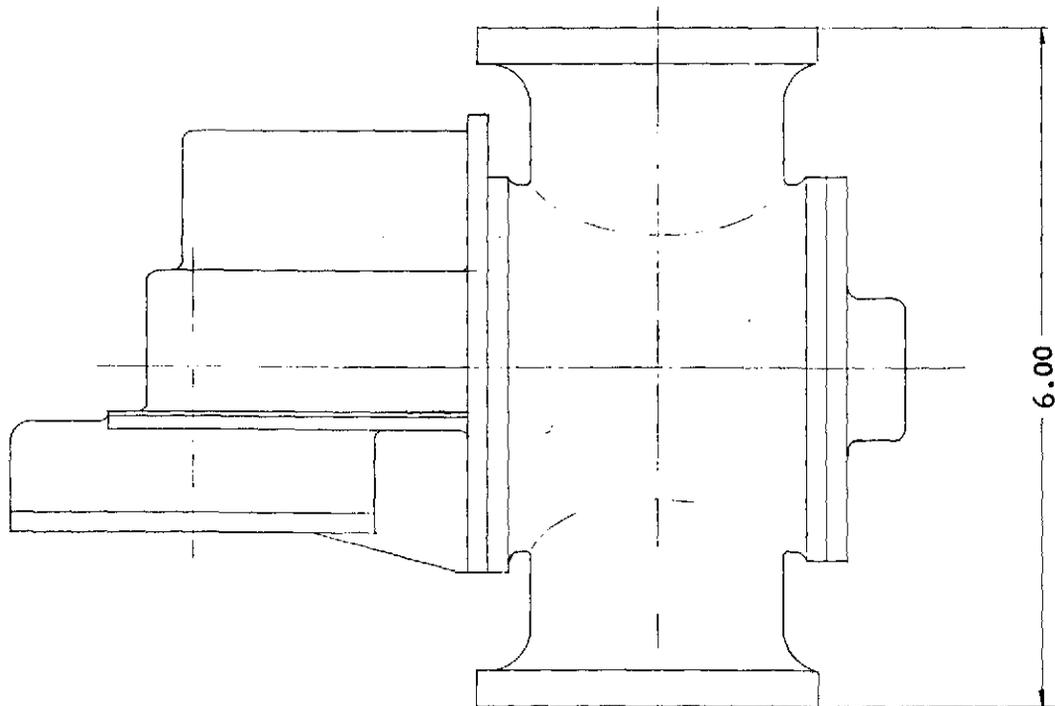
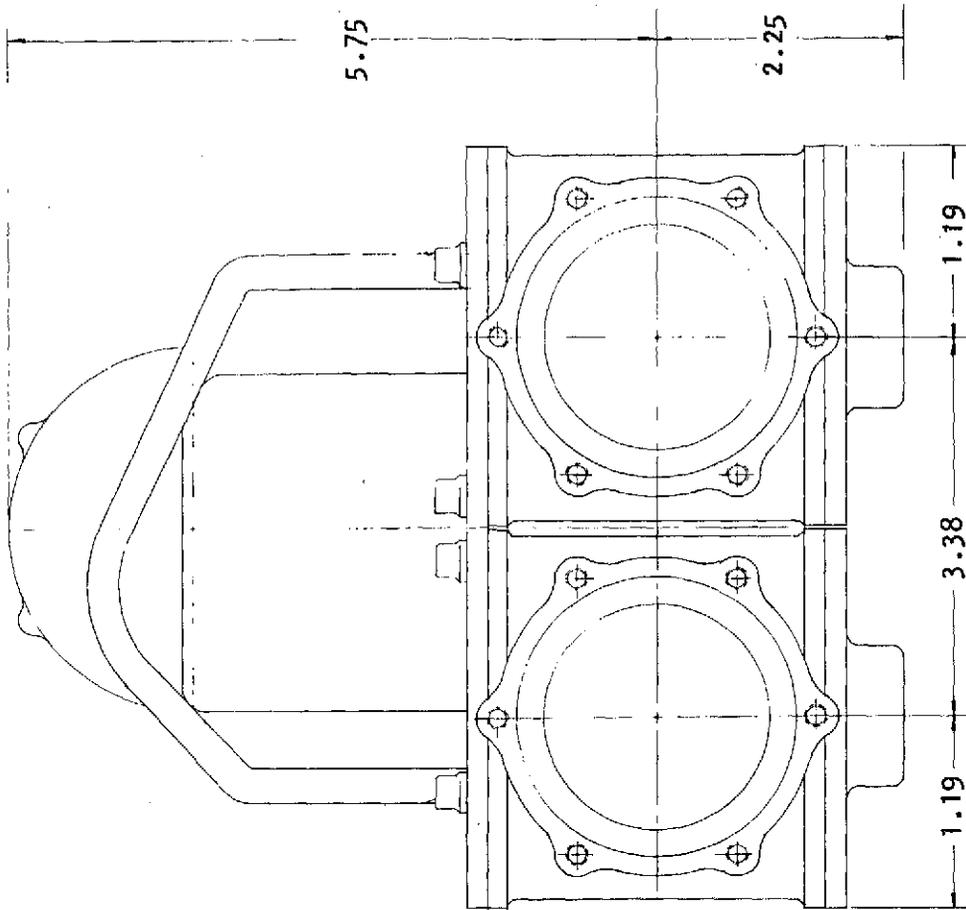
OME PROPELLANT SHUTOFF VALVE FINAL TRADEOFF

	Weighting Factor	Dual Poppet (Balanced)	Ball Valve (Solenoid seal liftoff)	Ball Valve (Internal driven seal liftoff)	Ball Valve (External driven seal liftoff)
Characteristic					
Weight	10	100	70	90	80
Power Consumption	8	72	64	80	80
Response Repeatability	6	48	60	60	60
Heat Rejection and Cooling	9	90	72	90	90
Maintainability	10	90	100	70	80
Purgeability	9	90	81	72	63
Reliability	10	100	100	90	70
Contamination Sensitivity	10	100	90	100	100
Vibration Sensitivity	7	56	56	63	63
EMI	5	50	40	50	50
Packaging	6	60	60	54	48
Development Risk (Pressure drop, thermal)	8	64	80	64	72
Cost (Development)	8	64	80	72	72
Cost (Unit)	7	70	70	63	56
TOTAL		1054	1023	1018	984

could be lowered by 3.75 psi with a resultant valve weight increase of less than 22.5 pounds a net OMS system weight savings would result. In addition some consideration was being given to possibly utilizing a series redundant valve only in place of the quad redundant valve on the OMS engine. Since one valve in a series redundant valve would have to flow twice as much propellant as one valve in a quad redundant valve it was apparent that a larger valve would be required for the series redundant arrangement in order not to exceed the 5 psi pressure drop allocated for the engine valve.

The dual poppet valve was therefore scaled up to achieve a pressure drop of 1.25 psid rather than the 5 psid required originally. Figure 4-32 is an envelope drawing of the scaled up bipropellant valve. This figure may be compared to Figure 4-33 which presents the quad redundant valve package featuring the five psid dual poppet valve. It is apparent from these layouts that the overall height of the valve does not change appreciably however the width of the bipropellant valve increases from 4.93 inches to 5.75 inches and the length of each bipropellant valve increases from 3.7 inches to 6.0 inches. A comparison of the performance characteristics of the series valve and quad valve packages is presented in Table 4-XIII. This comparison shows that the series valve package would be appreciably lighter although the valve would be somewhat slower and would require provisions for a holding power circuit in the electronic driver to prevent overheating of the torque motor. Total power consumption of the series valve package would actually be less than that of the quad valve package. On the other hand it would be possible to employ a quad valve package featuring a total pressure drop of only 1.25 psid by using two series valve packages in parallel and this configuration would then result in a quad valve weight of approximately 25 pounds or a net weight increase of 8 pounds. Consequently by applying the tradeoff factor of 6 pounds of weight for every one psi of pressure drop a total OMS weight savings of 14.5 pounds could be realized. It was this conclusion that caused NASA technical personnel to redirect the program to perform the detailed design of the dual poppet valve for the configuration featuring the 1.25 psid pressure drop.

At the same time an interest developed for the preparation of the detail design of a ball valve since all of the valves employed on rocket engines of similar size during the Apollo program were ball valves and since it was felt that there was some risk in achieving the pressure drop characteristics with the dual poppet valve because of the rather complex flow path through the valve. Consequently a weight comparison of the ball valves between the 5 psid and 1.25 psid pressure drop was also made. The results of this comparison as well as the results of the dual poppet valve comparison are presented in Figure 4-34. As evident from this figure the predicted weight for the poppet valve and the ball valve at 1.25 psid pressure drop requirement are nearly identical. It is also apparent from this curve that if even lower pressure drop requirements are of interest the ball valve design definitely is more attractive from a weight point of view. Since the weight characteristics of the ball valve were quite attractive at the low pressure drop the program was then therefore redirected to detail design both the poppet valve and the ball valve for this pressure drop.



SERIES REDUNDANT VALVE DESIGN LAYOUT

4-56

Figure 4-32

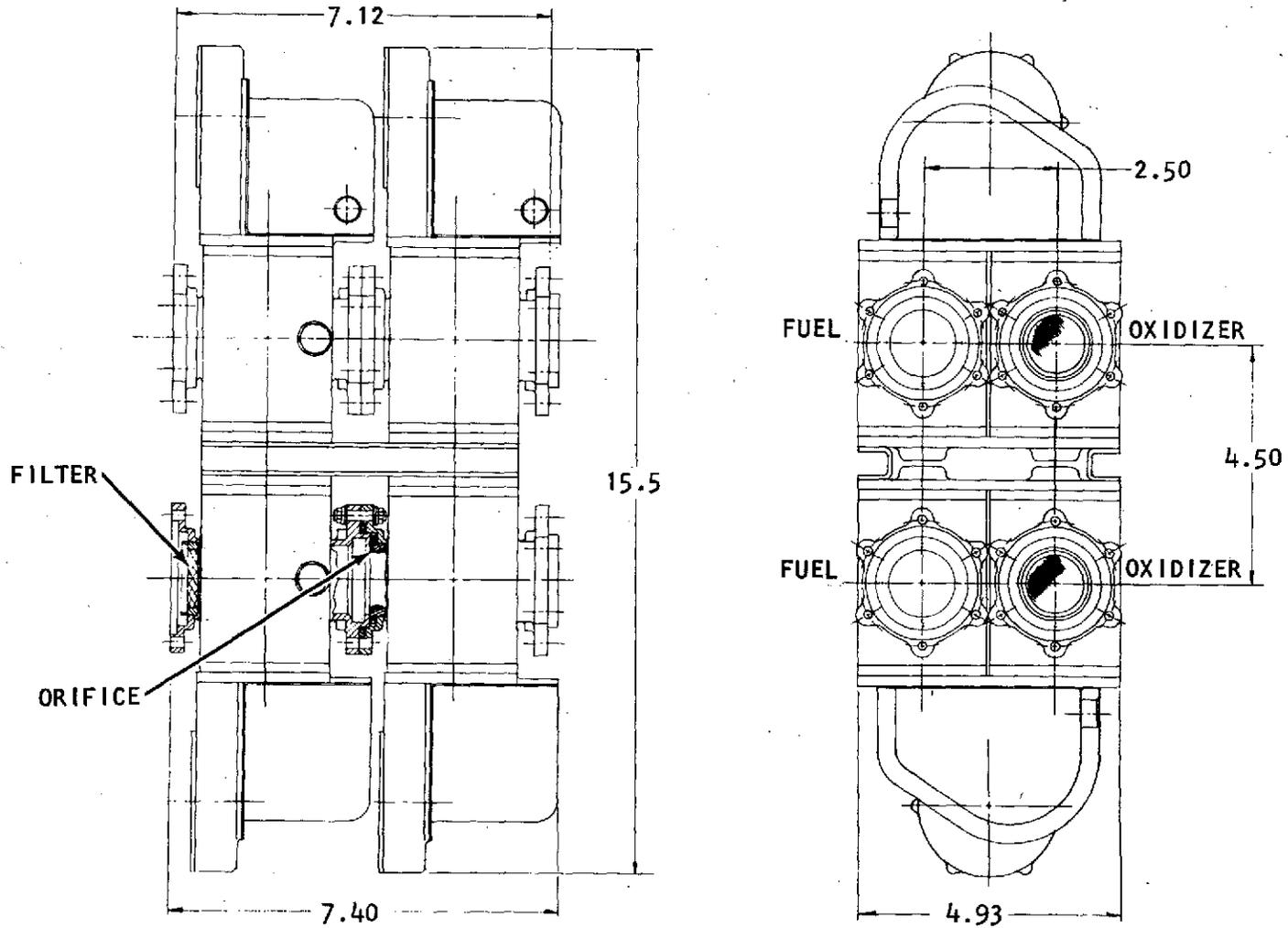


Figure 4-33

QUADREDUNDANT VALVE DESIGN LAYOUT

TABLE 4-XIII
 DUAL POPPET VALVE COMPARISON
 QUAD VALVE PACKAGE VS. SERIES VALVE PACKAGE

	QUADVALVE PACKAGE	SERIES VALVE PACKAGE
WEIGHT (LBS.)	17.0	12.3
• POWER CONSUMPTION (WATTS)	160	130/80*
• RESPONSE (MS)		
OPEN	300	325
CLOSE	250	290
• PRESSURE DROP (PSI) AT RATED FLOW	5	5

*HOLDING POWER

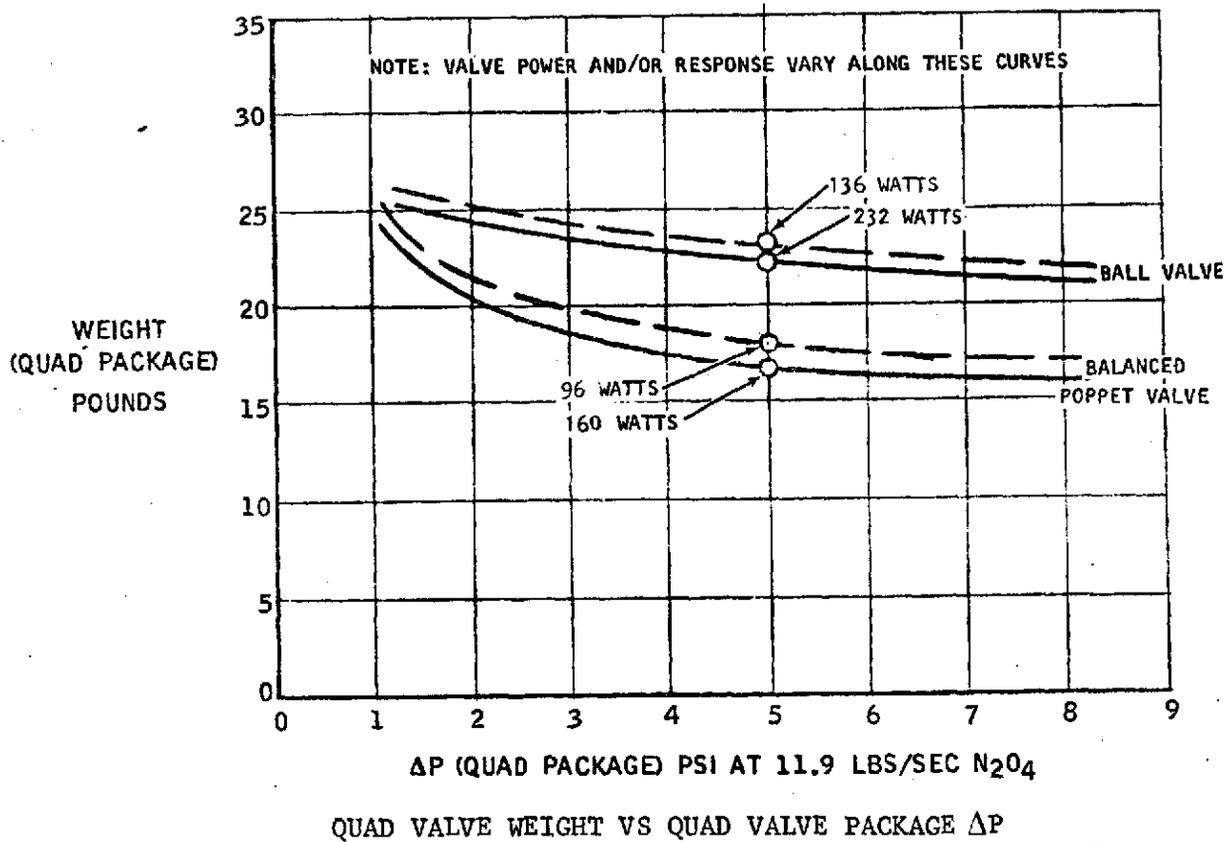


Figure 4-34

5.0 DETAIL DESIGN

The detail design effort performed during this program included the complete detail design of a dual poppet bipropellant valve and a ball bipropellant valve as required in the quad redundant valve package. The pressure drop requirement for each of the quad redundant valves was 1.25 psid at the nominal oxidizer flow rate of 11.9 pounds per second of nitrogen tetroxide. This pressure drop was allocated to the valve inlet and outlet manifolding, inlet filter, and bipropellant valves at 0.25, 0.2, and 0.8 psid respectively. In other words the pressure drop requirement through one side of the bipropellant valve was 0.4 psid at a flow rate of 6 pounds per second of nitrogen tetroxide.

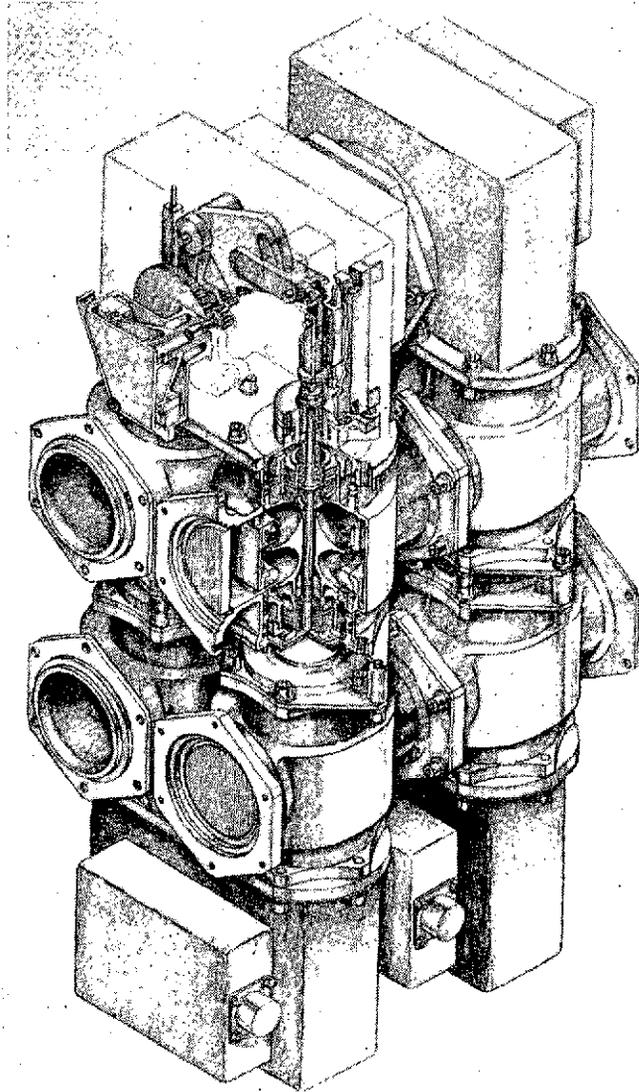
Since the detail analysis performed in support of the valve tradeoff study presented in the preceding section was based on a quad valve pressure drop requirement of 5 psid substantial additional analysis had to be performed to scale up the valve designs previously analyzed. Consequently final valve design criteria and sizing analysis for the lower pressure drop configuration are presented in this section. Also the final predicted valve performance characteristics are described herein.

To substantiate that the flow passage through the dual poppet valve was capable of meeting the required pressure drop prior to the initiation of the fabrication of the complete dual poppet valve NASA directed the Marquardt Company to also design a flow fixture which simulated this flow passage. This fixture was therefore also detail designed and is described in this section of the report.

5.1 DUAL POPPET VALVE

An artists conception of the dual poppet valve that was detailed for the OMS engine is shown in Figure 5-1. The valve is shown in the quadredundant arrangement featuring fuel and oxidizer flow in opposite directions. The two ports shown at the far left of the Figure are the outlet ports of the two parallel legs of the oxidizer valve and the two ports which feature an inlet screen are the inlet ports of the parallel legs of the fuel valve. As shown in this view, the parallel legs of either the oxidizer valve or the fuel valve have not been combined to a common oxidizer or fuel inlet or a common oxidizer or fuel outlet; it was determined that the best design approach would be pursued by addressing the routing of the ducting to a common point as part of the valve to engine integration study which is performed by the engine supplier. As evident from Figure 5-1, the quadredundant valve consists of four clearly distinguishable bipropellant valves. The quadredundant valve was designed so that one bipropellant valve could readily be removed from the package. Since the bipropellant valve is a complete unit by itself, the design and development efforts specifically addressed the bipropellant valve rather than the quadredundant valve package.

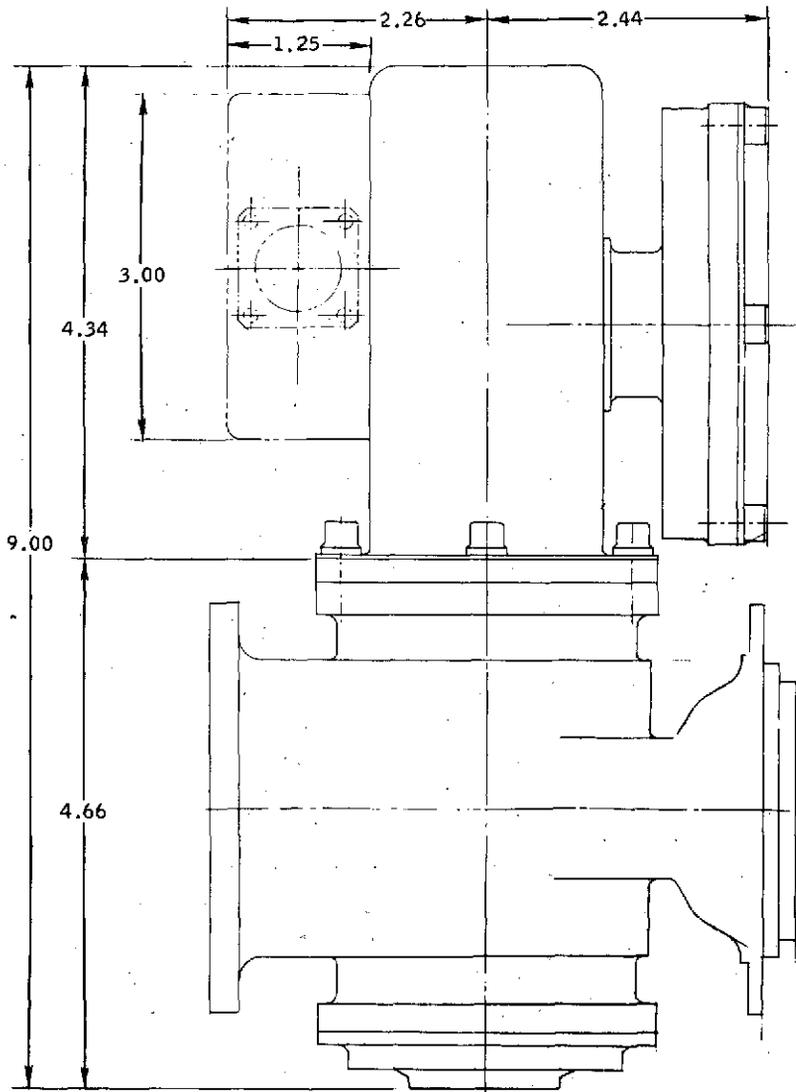
An elevation view of one bipropellant valve is shown in Figure 5-2. In this view, it is easy to recognize the major components of the valve. The top half of the view shows



NEG. 74-356-11

OME POPPET VALVE

Figure 5-1



DUAL POPPET VALVE
ELEVATION VIEW

Figure 5-2

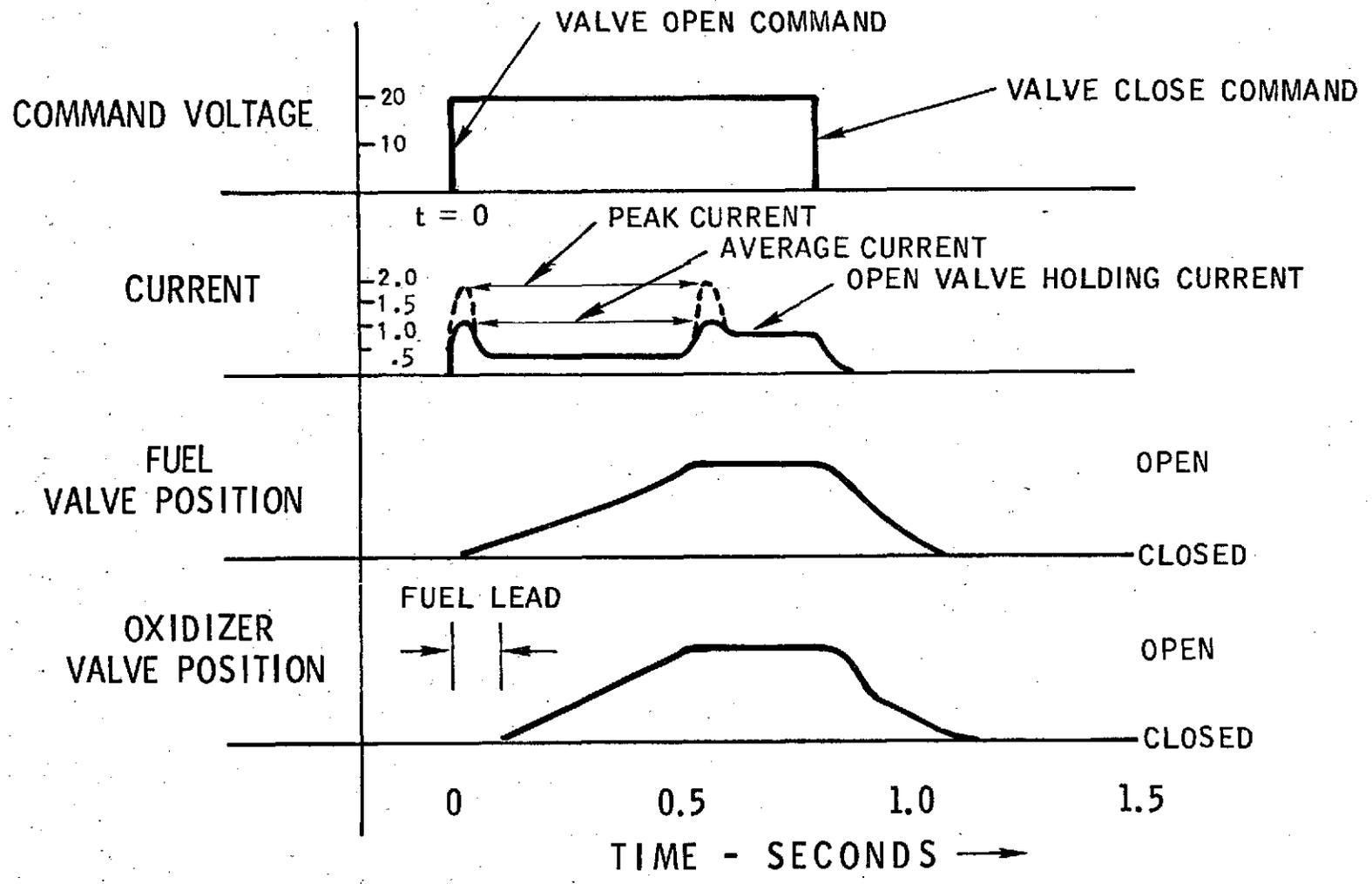
the actuator which is comprised of the electronic package on the left side, the gear box in the center, and the brushless DC torque motor on the right side. In the bottom half of the view the valve housing is evident; the second valve housing of the bipropellant valve is directly behind this valve housing and is therefore hidden in this view. Figure 5-2 also shows the overall dimensions of the bipropellant valve. Perpendicular to this view the maximum dimension is 6.75 inches.

Valve Operational Description

Details of the OME dual poppet valve are evident from Figure 5-1. The valve is called a dual poppet valve because it features two poppets in each valve mechanism. These two poppets are in parallel flow paths and are utilized to achieve pressure balancing. Thus, the differential pressure forces acting on one poppet oppose the differential pressure forces acting on the other poppet except for a small amount of pressure unbalancing which is intentionally included to provide the necessary sealing closure forces. Utilization of pressure balancing in this poppet valve results in the requirement for relatively low actuator forces and permits the all-electric torque motor actuation. The actuator includes a brushless D. C. torque motor which, through a two-stage, 36:1 gear reduction, drives an adjustable linkage which, in turn, via two pivot points, pushes on the poppet shafts of the oxidizer and fuel valves to open them. The adjustable linkage permits various fuel leads during the opening motion. An electronic package required for commutating the brushless D. C. torque motor is located on the side of the actuator housing as evident in Figure 5-2.

A typical actuation sequence is shown in Figure 5-3. To open the dual poppet bipropellant valve, a nominally 28-volt D. C. electrical signal is applied to the electronic package. This electronic package, in turn, drives the torque motor which, through the gear train, overcomes the pressure unbalance and spring forces in the valve mechanisms and opens the poppets. Fuel lead characteristics are illustrated in Figure 5-4. A stop located in the actuator, in combination with an impact absorbing spring, stops the valve motion in the open position. The stop in the actuator is adjusted such that the motor stops with only one winding being energized (See Figure 4-17 for the sequence of alternately driving each winding in the two phase motor) to reduced the holding power by 50%. When the valve is to be closed, the electrical signal to the electronic package is terminated. This, in turn, permits the closing spring and axial guidance flexures located in each valve mechanism to drive the actuation mechanism backwards. During the closing motion, the closing velocity is limited by dissipating the energy generated by the torque motor in its windings. Thus, closing impact loads are minimized.

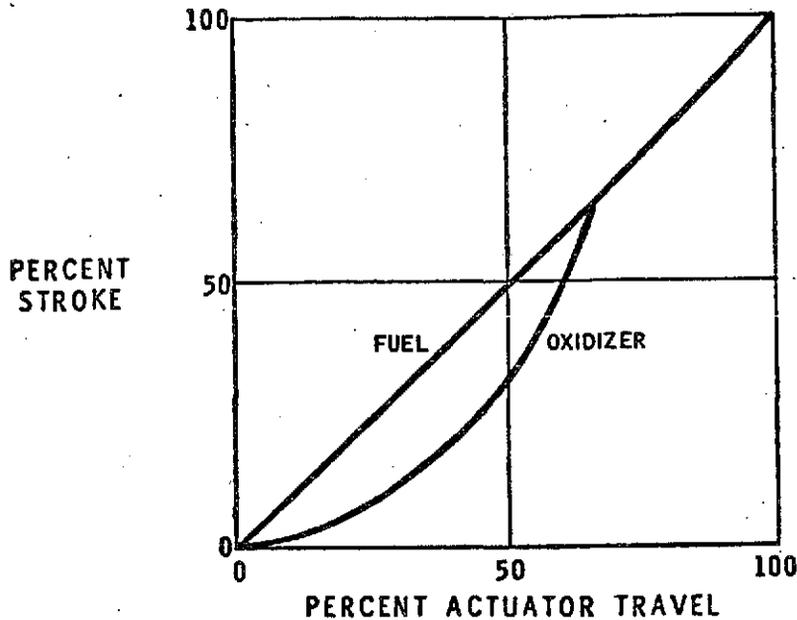
The total opening stroke is 0.20 inch. The inlets and outlets of the dual poppet valve are in line. Flow coming into the valve splits and flows through two spider arrangements. The flow then makes an approximately 180° turn to flow into a centrally located cavity from where it is expanded to the downstream flange. Details of construction of the flow path, valve mechanism and actuator are presented in the next paragraphs.



POPPET VALVE ACTUATION SEQUENCE

5-5

Figure 5-3



LEAD-LAG CAPABILITY
 BASELINE REQUIREMENT - 100 MS FUEL LEAD
 POPPET VALVE OPENING MOTION

Figure 5-4

Construction Details and Materials of Construction

Valve Mechanism:

The two poppet/seat interfaces contained in each valve mechanism are identical in concept although they differ slightly in dimension and arrangement because of opposing directions of flow. The poppet/seat interface features a well lapped, flat poppet surface mating with a 0.030 inch wide teflon (TFE) seat. The teflon is retained in the seat such that a metal backup exists immediately downstream of the teflon to prevent cold flow. The effective seating diameter of the seats at both poppets and the effective diameter of the single convolution bellows at the upper seat are matched except for the differential area required to establish a seating stress at both seats. This same pressure differential force can also act to relieve pressure through the seat to the upstream side of the valve in the event the downstream pressure exceeds the upstream pressure by more than 25 psi. Such a condition is conceivable in the quadredundant valve package due to the thermal soakback from the engine.

The valve mechanism of the dual poppet valve is completely friction free. There are no parts to rub or wear or to generate contamination. Axial guidance of the poppets is accomplished by means of metallic guidance flexures located at each end of the poppet stem. The egress of the poppet stem through the valve housing is sealed by means of a redundant, hydroformed bellows assembly. A test port between the two parts of the redundant bellows assembly permits leak checking of each of these bellows. This test port is normally capped during operation.

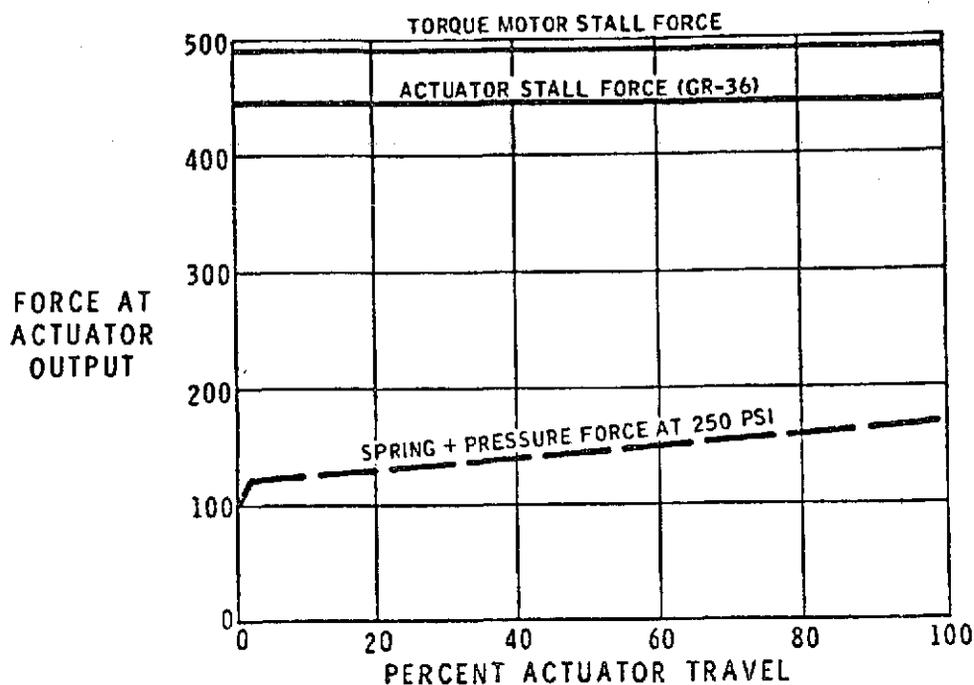
The dual poppet valve mechanism has been designed to readily permit complete disassembly of the valve and to permit the inspection, removal and replacement of such critical items as the seats, the bellows shaft seal, poppets, and axial guidance flexures. Materials of construction of the valve mechanism include a housing assembly made of titanium 6AL-4V for the oxidizer side and aluminum 6061 for the fuel side. Other titanium or aluminum parts include the end closure, flexure retainer nut, both poppets, poppet nut, and the flow orifice. Both the shaft seal bellows and the seat bellows are made from Inco 718, heat treated to Rockwell R_C 32 to 36. Inconel 600 is used for such parts as the seat stop, the seat, retainer, flow baffle, bellows shaft adapter, bellows end flange, bellows vent sleeve, and the spacers between the flexure plates. The axial guidance flexure plates are also made of Inco 718 and the flexure assemblies are brazed together. The bellows leak test tube is made of type 321 stainless steel and the return spring of type 302 stainless steel. All seals employed in the valve mechanism are of the teflon jacketed type.

To prevent galvanic corrosion between the aluminum parts and the other materials employed in the construction of the valve, the aluminum is impregnated with teflon in accordance with the Nituff or Tuftram methods. Otherwise, all of the valve materials are fully propellant compatible and no platings or other protective coatings are required. With the exception of the teflon utilized in the valve, the valve is of an all metallic construction and is therefore suitable for a service life of ten years without the need for any type of age control.

Actuator:

The actuator consists of a centrally located aluminum structure which supports the first stage reduction gear as well as the second stage sector gear and adjustable linkage. In addition, the pivot pins required for transmission of the stroke to each of the poppet push rods as well as the guidance for these push rods are contained within this central aluminum structure. The aluminum structure also serves to support the position indicators located at each of the valve push rods and the position indicator electrical connector. Surrounding all of the parts just described is a box-like aluminum case with a honeycomb stiffened backface to support the electric torque motor. Opposite the torque motor another aluminum case is attached which houses the electronic package. (See Figure 5-2). Inside the aluminum case at one end, a 15-pin electrical connector is located which routes all of the wiring from the electrical torque motor to the electronic package. All of the components of the actuation mechanism are made of teflon impregnated 6061 aluminum with the exception of the poppet push rods which are made from custom 455, the carrier screws and dowels which are made from 300 series stainless steel, electrical wiring, electronic components, magnetic materials in the torque motor and the windings of the torque motor. The teflon impregnated 6061 aluminum is utilized at load points throughout the actuator drive train, such as bearings surfaces, gears, and pivot pins. This material has been proven in a number of space applications to be an excellent self-lubricated bearing surface.

The dual poppet valve actuation force requirements are presented graphically in Figure 5-5. As illustrated in this figure there is ample margin between the required



POPPET VALVE ACTUATION FORCES
BIPROPELLANT VALVE

Figure 5-5

actuation forces and the actuation forces available from the torque motor and gear box. During the analysis effort to scale up the poppet valve and the ball valve to the lower pressure drop requirement it was decided to size the actuators for both of these valves such that the same torque motor could be utilized in either valve. A specification for this torque motor was subsequently prepared and is presented in Table 5-I.

TABLE 5-I
BRUSHLESS DC TORQUE MOTOR SPECIFICATION

Performance Requirements:

- Peak torque (stall) - 65 oz in
- No load speed - 75 rad/sec. @ rated voltage
- Inner member rotates
- Number of phases - 2

Mechanical Requirements:

- Number of poles - 14
- Max. winding temp. - 130°C
- Corrosion resistant materials or coatings to be used throughout
- OD - 3.625"

ID - 2.500"

(See Drawing
For all Dim.)

TABLE 5-I (Cont.)

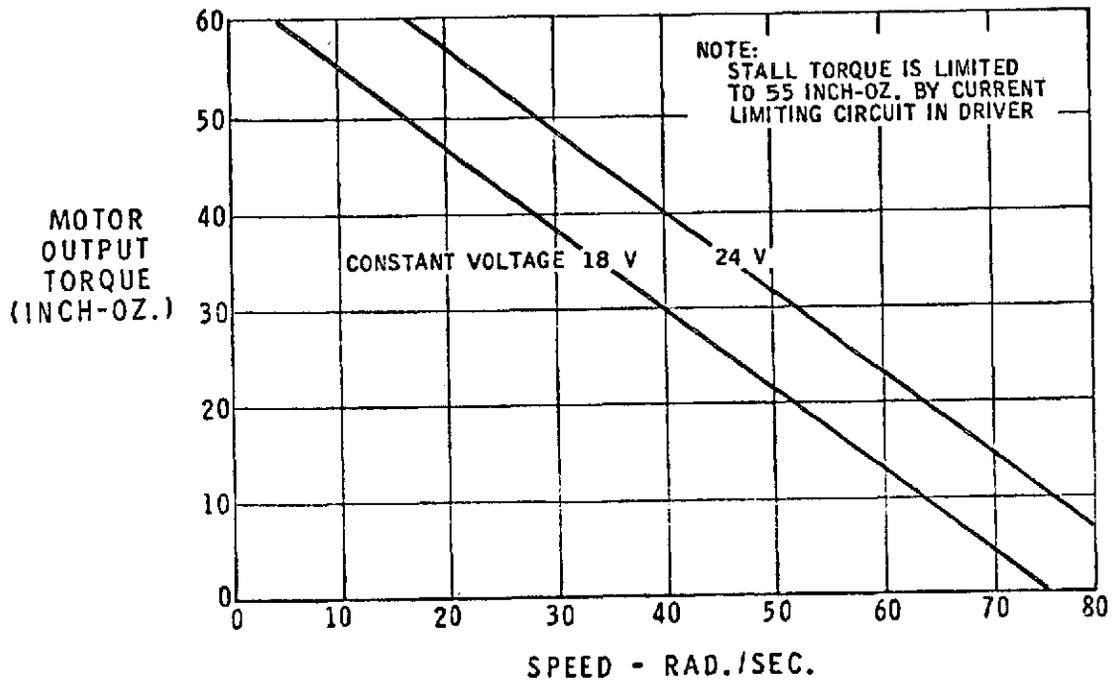
Electrical Requirements:

- Max. Power Per Phase - 20 Watts
- Rated Voltage @ 25°C - 18V
- Current at Rated Torque - 1.1 amp/phase
- Inductance - 6 millihenries
- Type Commutation - 2 Hall Generators (FW Bell Model FH-300)

A brushless DC torque motor including Hall effect devices was subsequently procured from a supplier. The electronic driver required to operate the brushless DC torque motor was designed at the Marquardt Company.

A digital electronic driver capable of supplying signals to the two motor windings in accordance with the control logic shown in Figure 4-16 was designed. The circuit design was made for a breadboard driver to permit development of the driver and to facilitate the adjustment of a number of driver constants. Specifically the driver permits a variation of a duty cycle from the nominal 60% to as much as 100% and to as little as 20%. To control the impact loads that occur in the drive train when the valve mechanism runs against the stops during the closing motion the driver also includes a dynamic breaking circuit. This dynamic breaking action is initiated by an adjustable timer circuit and the amount of breaking can also be varied by varying the back EMF to the winding. To prevent the possibility of demagnetization of the torque motor rotor by driving the windings with too much power an adjustable current limiting circuit was also included in the driver. Thus the maximum current going to the windings can be preset. The driver was designed to operate over a voltage range of 18 to 36 volts and included its own power supply to establish 15 and 7.5 volt reference voltages as required at various junctions in the circuit. All of the components utilized in the electronic driver were solid state components such that the power consumption of the driver was limited to 8 watts. Since the peak power required by the torque motor was 40 watts the total peak power required by a bipropellant valve was 48 watts, the average power per bipropellant valve using a 60% duty cycle was 32 watts, and the power required to hold the valve open was 28 watts.

The opening response of the poppet valve is primarily a function of the velocity of the electric torque motor over the required stroke. The average velocity of the motor is computed from the torque speed characteristics of the motor assuming an average load torque. Figure 5-6 shows the torque speed curve of the selected torque motor as a function of applied voltage and Figure 5-5 shows the poppet valve actuation forces required as well as the actuator output stall force available. From Figure 5-5 the average load force (springless pressure force) is approximately 150 lbs. This corresponds to a torque motor output requirement of 20 inch-ounces. Therefore, from Figure 5-6 the average motor speed is 65 radians per second. The total opening speed is then computed as the sum of the motor torque rise time ($\tau = .030$ seconds) and the average velocity. The equation describing the opening time is:



TORQUE MOTOR CHARACTERISTICS
PART NUMBER X29580

Figure 5-6

$$t_o = T_m + \frac{\Delta X}{\dot{\theta}_{MAvg.}} \quad (GR)$$

$$t_o = 0.48 \text{ Sec.}$$

where:

T_m - Torque Motor Mechanical Time Constant (0.03 Sec.)

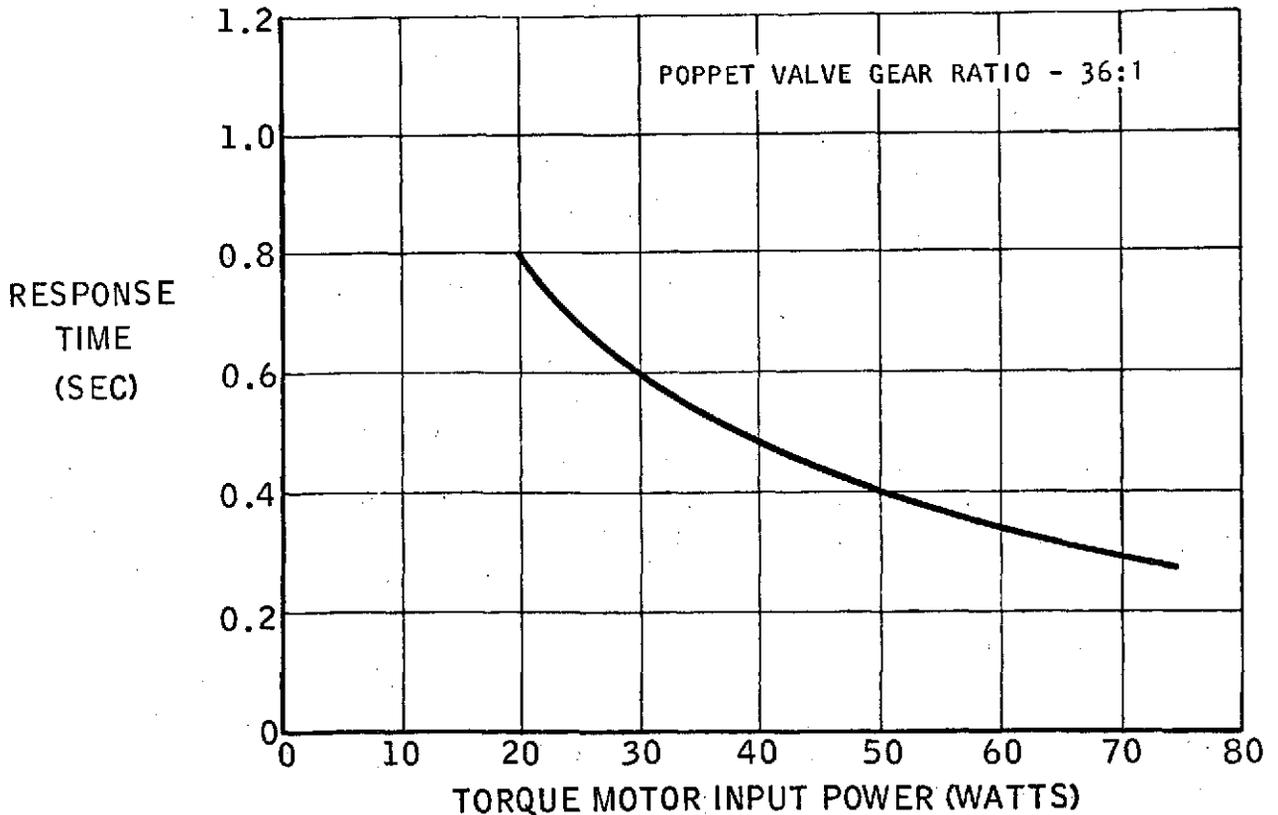
$\dot{\theta}_{MAvg.}$ - Average Velocity Over Valve Stroke

(GR) - Gear Ratio (36:1)

ΔX - Valve Stroke (0.2 In.)

r_p - Output Lever Arm for Poppet Valve (0.25 in.)

Opening response characteristics of the dual poppet valve were also determined as a function of torque motor operating power. This data is presented in Figure 5-7.



POPPET VALVE OPENING RESPONSE

Figure 5-7

As noted in the figure the response characteristics are at a constant gear ratio of 36:1. Thus in the event faster response characteristics are desired or slower response characteristics are permitted the resultant changes in electrical power requirements may be determined from this curve.

The closing response of the dual poppet valve is a function of the net closing torque (resulting from the spring forces in the valves minus the friction torque) and the inertia of the complete valve. In addition dynamic braking is employed to minimize closing time and to prevent excessive impact forces from occurring in the drive train when the closing position stop is reached. The dynamic braking employs the principal that by controlling the current generated by the torque motor when being driven as a generator a decelerating torque is obtained from the motor coils. Consequently the motor can be "free wheeled" accelerated up to a high speed and then remain at that speed until impact on the mechanical stop. The equation describing the total closing time is:

$$t_c = \frac{I_m (GR) \dot{\theta} \text{Max.}}{r_p (F_s + F_p)} + \frac{\left[\Delta \dot{\theta} \text{Tot.} \cdot \frac{I_m (GR) \dot{\theta} \text{Max.}}{r_p (F_s + F_p)} \right]}{\dot{\theta} \text{Max.}}$$

$$t_c = 0.32 \text{ Sec.}$$

where:

- $\dot{\theta} \text{ Max.}$ - Maximum Closing Velocity of Motor at Impact (100 Rad./Sec.)
- (GR) - Gear Ratio (36:1)
- I_m - Torque Motor Inertia (6.7×10^{-4} in. oz. sec.²)
- r_p - Output Lever Arm for Poppet Valve (0.25 in.)
- $(F_s + F_p)$ - Closing Force
- $\Delta \dot{\theta} \text{ tot.}$ - Total Travel of Motor

The dual poppet valve was stress analyzed to feature a 1.5 safety factor on proof pressure and a 2.0 safety factor on burst pressure. Minimum allowable wall thicknesses were set at 0.030 inches to enhance manufacturing methods and to minimize potential deflection problems in the valve. A particularly important part of the stress analysis was the determination of maximum stresses in the drive train especially when the motion of the drive train was terminated by the open position stop or the closed position stop. During these analyses it was determined that the open position and closed position stops had to be reasonably compliant to minimize impact stresses. Consequently two leaf spring type stops were incorporated into the poppet valve. With these stops it was possible to limit the maximum contact stress in the actuator gears to 65,000 psi and the maximum tooth bending stress to 19,000 psi. This compares to typical operational contact stresses in the gears of 43,000 psi

and typical operational tooth bending stresses of 9,000 psi. The resultant safety margins at maximum stress were determined to be 0.21 and 2, respectively, for a cycle life of 1 million cycles. The material of construction of the gears was 7075-T 6 aluminum with nituff as mentioned previously.

Based on the design layout of the dual poppet bipropellant valve a detailed weight estimate was prepared. A summary of this estimate is presented in Table 5-II.

TABLE 5-II

POPPET VALVE WEIGHT

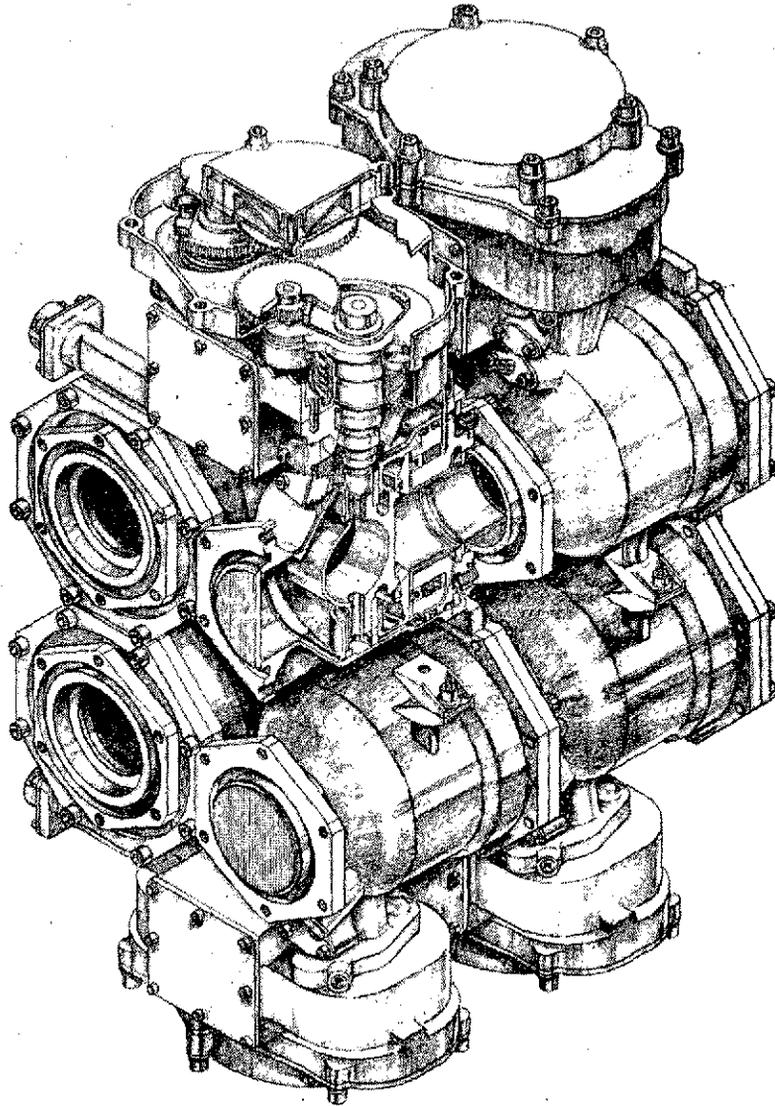
VALVE COMPONENTS	WEIGHT - POUNDS		
	Al	Ti	Cres
HOUSING	0.218	(0.357)	
INLET FLANGE	0.203	(0.333)	
OUTLET FLANGE	0.141	(0.230)	
END CLOSURE	0.121	(0.198)	
FLEX SUPPORT	0.062	(0.102)	
SEAT ASSEMBLY FIXED			0.139
SEAT ASSEMBLY FLOAT			0.246
FLEX SUPPORT	0.061	(0.101)	
FLEX RETENSION NUT (2)	0.055	(0.089)	
POPPET FLEX SEAT			0.213
POPPET FIXED SEAT			0.106
POPPET SPRING			0.002
POPPET NUT			0.038
POPPET SHAFT			0.074
BELLOWS VENT FLANGE			0.103
BELLOWS (2)			0.089
FLEX SUPPORT (INNER)	0.010	(0.016)	
FLEX ASSEMBLY (2)			0.192
SHIM	0.010	(0.016)	0.029
WAVE SPRING			

NOTE: Brackets indicate Oxidizer Valve.

ACTUATOR COMPONENTS	WEIGHT (LBS.)
TORQUE MOTOR	0.610
TORQUE MOTOR COVER	0.107
TORQUE MOTOR HOUSING	0.194
ROTOR SHAFT	0.056
QUILL	0.008
DRIVE PINION	0.004
DRIVE GEAR	0.014
SECTOR GEAR	0.120
OXIDIZER CAM LINK	0.004
TAPPET LINK	0.007
TAPPET CARRIER	0.014
TAPPET (STEEL)	0.020
HOUSING	0.490
ELECTRONICS	0.250

- FUEL VALVE ————— 2.099 LBS.
- OXIDIZER VALVE ————— 2.659 LBS.
- ACTUATOR ————— 1.911 LBS.
- TOTAL BIROPELLANT VALVE 6.659 LBS.
- QUAD PACKAGE WEIGHT 26.6 LBS.

As evident from Table 5-II a total bipropellant valve weight of 6.659 lbs. was predicted and this corresponds to a quad valve weight of 26.6 lbs. The determination of this weight essentially confirmed the weight predictions made earlier during the preliminary valve trade-off studies that resulted in the selection of the electrically operated dual poppet valve as the best concept for the OME valve application.



NEG. 74-356-12

OME BALL VALVE

Figure 5-8

The final predicted valve performance characteristics of the quadredundant dual poppet valve are listed in Table 5-III. It should be noted that the pressure drop characteristics of the valve at 1.4 psi is made up of a 0.2 psi pressure drop for the inlet screen

TABLE 5-III
PERFORMANCE CHARACTERISTICS QUADREDUNDANT DUAL POPPET VALVE

Pressure Drop	- 1.4 psi @ 11.9 lbs/sec N ₂ O ₄ (for inlet and outlet manifolding add 0.25 psi)
Operating Pressure	- 250 psi nominal
Opening Response	- 480 ms @ 250 psi inlet pressure, 24-30 volts DC, 40-200°F
Closing Response	- 320 ms @ 250 psi inlet pressure, 11.9 lbs/sec N ₂ O ₄ , 24-30 volts DC, 40-200°F.
Internal Leakage	- 10 SCC/Hr Helium maximum @ 2-250 psi inlet pressure
External Leakage	- 1.66 x 10 ⁻⁷ SCC/sec Helium maximum per joint @ 2-250 psi inlet pressure
Operating Voltage	- 24-32 VDC
Electrical Power	- 192 Watts to open, 96 Watts to hold open
Operating Temperature	- 40-200°F
Cycle Life	- 4000 wet, 6000 dry
Service Life	- 500 missions or 10 years maintenance free
Weight	- 27 lbs
Acceleration	- 4 g's maximum
Compatibility	- N ₂ O ₄ and MMH in combination with their combustion products moisture, and salt spray

and a 0.6 psi pressure drop for each bipropellant valve at 1/2 the flow rate. Thus the original pressure drop goal of 1.25 psid was exceeded somewhat. The reason that the pressure drop is slightly higher is due to the fact that the manifolding inside each valve body from the inlet to the poppet seat interface in the final design review was determined to be somewhat more restrictive than originally intended. However, it was concluded that this slightly higher pressure drop did not warrant redesign of the valve housing and its associated envelope and weight increase. The remaining performance characteristics listed in Table 5-III show that the performance requirements listed in Section 3 have been met and that a relatively low weight valve configuration has been achieved.

5.2 BALL VALVE

An artist's conception of the quadredundant ball valve is shown in Figure 5-8. As in the case of the poppet valve, this valve is arranged for counterflow of the oxidizer and the fuel. Thus, the two ports on the left side of the Figure are the outlets of the two parallel oxidizer legs of the valve and the two ports featuring the inlet screens are the inlets of the

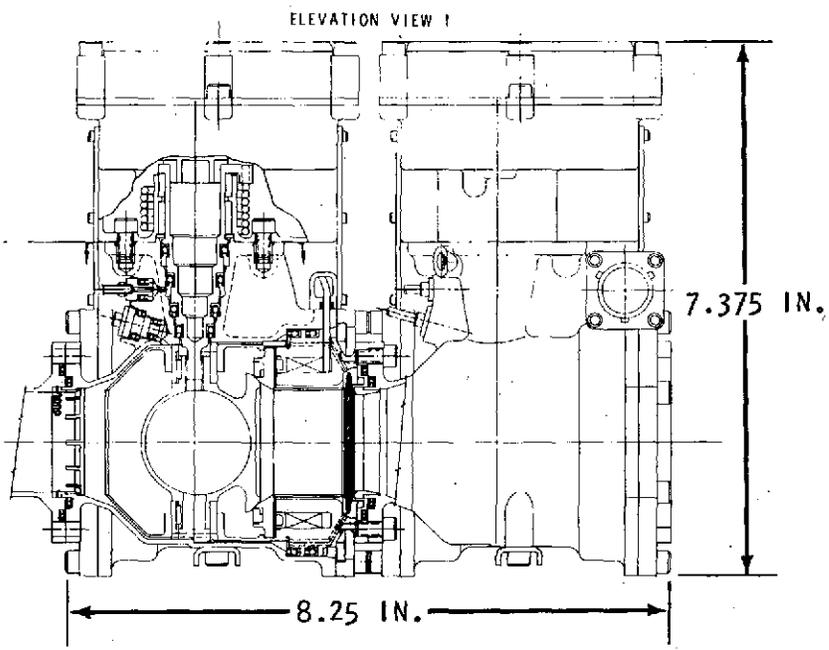
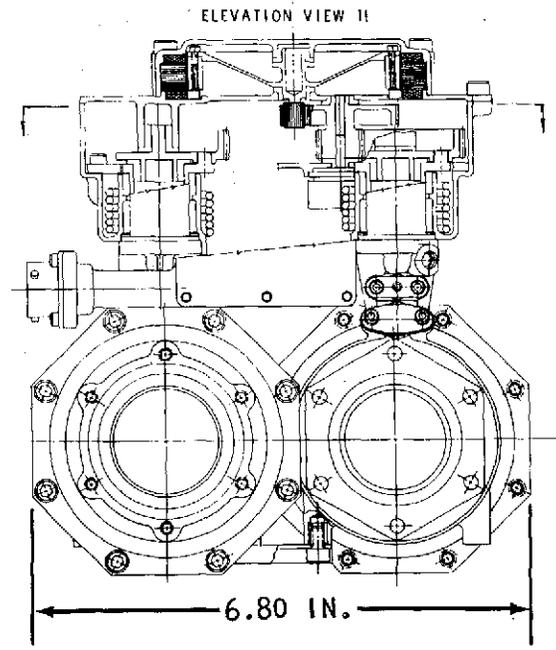
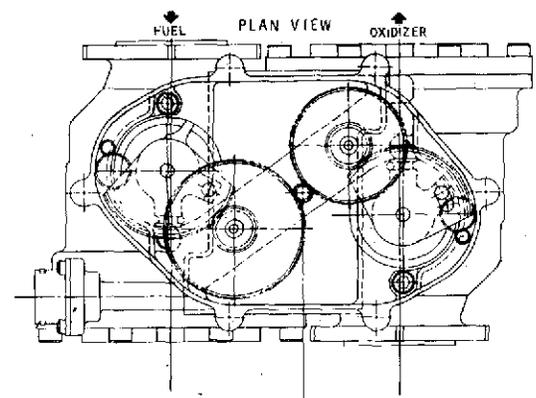
two parallel legs of the fuel circuit. The quadredundant ball valve is also made up of four independent bipropellant valves each featuring a fuel valve, an oxidizer valve and a common actuator. The electronic driver of the brushless DC torque motor used to drive the actuator mechanism is located below the gear box and between the shafts to the fuel and oxidizer valves in each of the bipropellant valves. Coverplates which close off this section of the actuator housing are readily visible in Figure 5-8 between the fuel valve, the oxidizer valve, and the gear box of the top bipropellant valve. As in the case of the poppet valve, the detail design and development efforts were concentrated on the bipropellant valve as the basic unit. A description of the valve follows.

5.2.1 Valve Operational Description

Cross sections of the actuator and valve mechanism of the ball valve configuration are shown in Figure 5-9. As evident from this figure, the ball valve employs a scalloped ball in combination with a retractable teflon seat to achieve long life and reliable sealing. The seat is mounted on a single convolution bellows which is deflected when the solenoid actuator is energized to retract the seat from the ball. The sealing closure interface forces are achieved by a combination of spring force available from the precompression of the single convolution bellows and an intentional pressure unbalance achieved by making the effective seating diameter of the seat slightly larger than the effective diameter of the single convolution bellows.

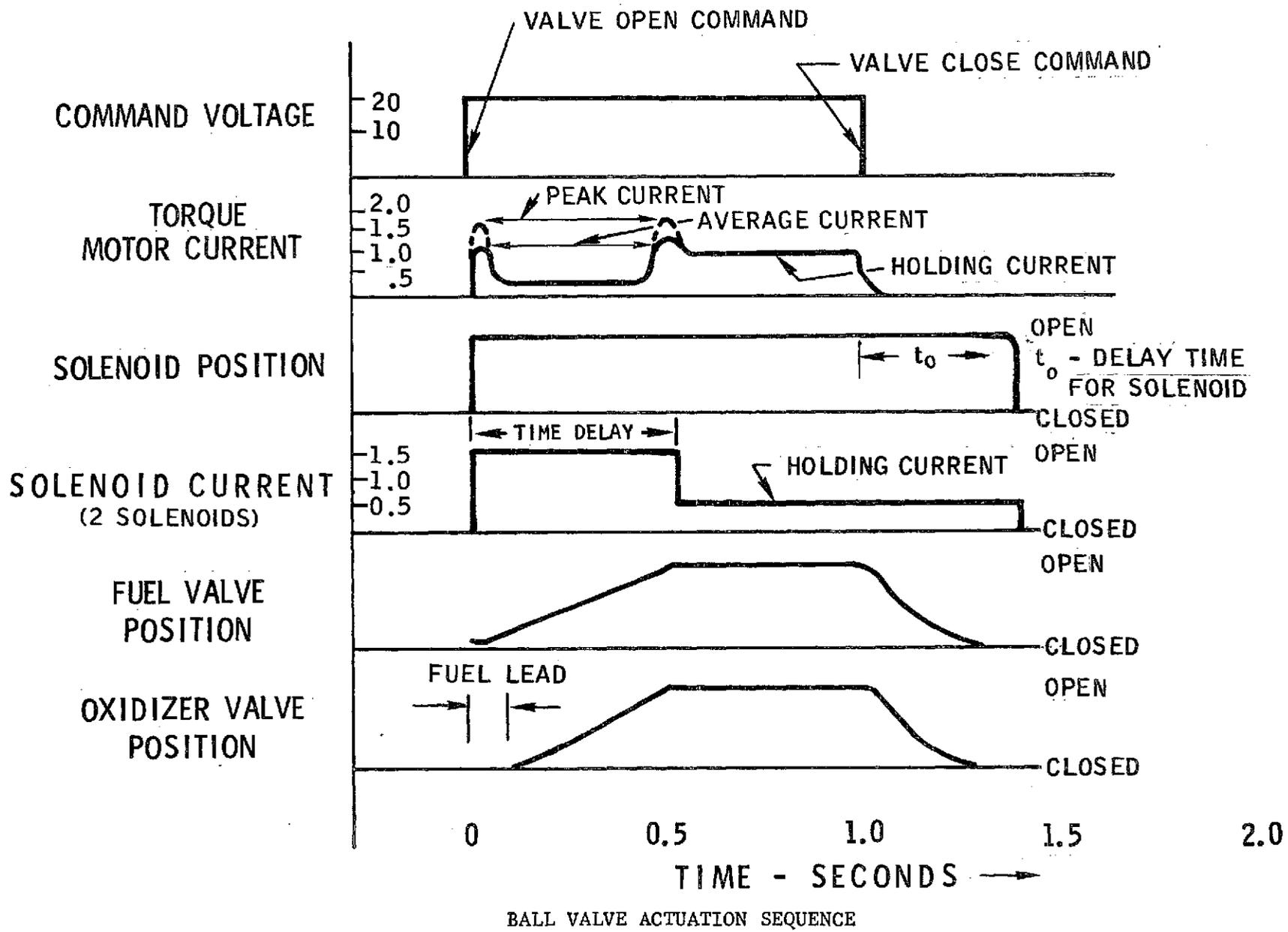
The shaft that turns the scalloped ball is extended through redundant seals into the actuator cavity. Inside the actuator cavity both the fuel valve shaft and the oxidizer valve shaft are linked through two stages of gears to the brushless DC torque motor which operates the bipropellant valve. The gear linkage includes slightly different gear ratios for the fuel and oxidizer valves in combination with a pin and slot arrangement on the oxidizer valve to permit valve fuel leads of up to 30%.

The ball valve actuation sequence is illustrated in Figure 5-10. To open the bipropellant valve a nominally 28-volt DC signal is transmitted through the electrical connector to the electronic package located in the center of the bipropellant valve. This signal is then applied simultaneously to the torque motor and to the seat solenoids. Since the time constant of the torque motor is considerably longer than that of the solenoid, the valve seat will be fully retracted by the time torque motor motion is initiated. When the torque motor starts to move, the fuel valve ball also starts to rotate because of its direct gear linkage to the torque motor. The oxidizer ball remains in the closed position until the clockwise end of the slot visible in the top right-hand view of Figure 5-9 catches up with the pin of the valve shaft. At that time, the oxidizer ball valve also starts to move. Due to the differences in gear ratio of the fuel and oxidizer valves, the oxidizer valve rotates somewhat faster so that it reaches the open position at the same time as the fuel valve. Valve full-open time is achieved in approximately one-half second.



BALL VALVE CROSS SECTIONS

Figure 5-9



5-18

Figure 5-10

After the bipropellant valve has reached the open position, the electrical power is automatically reduced to a holding power level. To close the valve, the electrical holding power to the torque motor is turned off. The two torsional springs located at the valve shafts then drive the valve mechanism and actuator to the closed position. The oxidizer valve reaches the fully closed position first and the fuel valve somewhat later, as defined by the slot and pin arrangement in the oxidizer valve driving mechanism. During the closing motion valve speed is limited by sequencing the brushless DC torque motor to act as a dynamic brake to minimize impact stresses in the actuator gear train when the fuel valve shaft contacts its stop. All the time while the bipropellant valve is closing the seat solenoid remains energized with the holding power. At a preset time interval the solenoid is deenergized and the seat then contacts the ball approximately 10 milliseconds later. In this manner, it is assured that no scrubbing whatsoever between the ball and the seat will occur during the closing motion. The valve position of the fuel valve ball and the oxidizer valve ball may be monitored independently by means of the rotary potentiometers located on each of the driving gears of the two valve mechanisms.

5.2.2 Construction Details and Materials of Construction

Ball Valve Mechanism

The ball valve features a scalloped ball consisting of a tubular section with a disc on one side. The edges of the disc have been machined and lapped to a spherical surface to achieve the ball sealing surface. The tubular section is used for structural and flow streamline reasons. The scalloped ball is supported at the tubular section by means of thrust and radial bearings made of Armalon. The scalloped ball is rotated by means of a shaft extending from the actuator section into the valve mechanism and connecting to the ball by a spline. To seal the propellant cavity, three radial Teflon-jacketed seals are employed at the shaft. The first two of these seals are redundant seals as required for the Space Shuttle application and the third seal is used to create a cavity which is connected to a tube fitting to which a leak check fixture may be attached to verify the integrity of the redundant shaft seals. There is also a fourth seal on the actuator shaft which is intended as an actuator housing seal to prevent contamination from entering the actuator housing. The space between the third and fourth seal is vented overboard through a 25-micron absolute screen.

The sealing closure consists of the spherical ball surface mating with a conical Teflon seat. Metal backing is provided downstream of the Teflon seal to prevent cold flow of the Teflon. The base of the seat acts as the armature for the solenoid actuator. A cylindrical section extends from the base to a single convolution bellows located downstream from the solenoid actuator at the outlet of the valve. Sealing closure interface forces are provided by the precompression of the single convolution bellows. During normal operation, the seat is retracted 0.005 inch before any ball motion is initiated. The ball rotational axis is located slightly eccentric with respect to the seat such that the ball moves away from the seat as it opens. In the full open position the clearance between the ball and the seat is 0.060 inch. The valve mechanism also includes a stop for the

seat in the forward direction. Thus in the event a solenoid actuator failure should occur, scrubbing between the ball and the seat will occur only during the initial 8 degrees of rotation. At that point the seat will have reached the stops just mentioned such that it can no longer follow the ball and the ball will rotate freely to its full open position.

The primary material of construction of the oxidizer valve is titanium (6AL-4V) and that of the fuel valve is aluminum (6061). These materials offer excellent compatibility characteristics with the respective propellants and combinations of moisture and salt air. Specific valve components to be made from the materials just mentioned included the housing, inner liner and bushings. The scalloped ball is made from Custom 455, again a highly corrosion resistant material, to obtain a hard, wear-resistant sealing surface. The seat, bellows and solenoid retainer Belleville spring are made from Inco 718. Other materials of construction include Inco 600 for the seat seal retainer, the cylindrical part supporting the seat, dual weld transition rings, and nonmagnetic section of the solenoid actuator. Type 446 Stainless Steel, which is the most corrosion-resistant magnetic steel available, is used for the magnetic section of the solenoid. The solenoid windings are copper with Pyre-ML insulation and teflon insulated stranded copper lead wires.

The valve mechanism utilizes no coatings or platings whatsoever. All materials are fully compatible with the propellants and their combinations with moisture and salt air. The valves are made entirely of metals and Teflon, therefore requiring no age control for the Space Shuttle application. Except for the aluminum in the fuel valve, the metals utilized are fully compatible with one another with respect to galvanic corrosion. To prevent galvanic corrosion, all aluminum parts are anodized and teflon-impregnated using the Nituff or Tuftram processes.

Actuator

The actuator consists of one primary housing with three "lids". One of these lids encloses the brushless DC torque motor to the actuator housing, and the other two constitute end plates to the electronic package, the other sides of the electronic package housing being formed by the actuator gear housing. The torque motor drives both the fuel ball and the oxidizer ball through a two-stage gear train. The gear ratio for the fuel valve is 21.4, the gear ratio for the oxidizer valve is somewhat greater, depending upon the exact amount of fuel lead desired. Torsional return springs are incorporated around each of the actuator shafts to accomplish the fail-safe closed requirement. The arrangement of gears, torque motor, return springs, position indicators, and electronic package are evident from the right-hand view of Figure 5-9. The last stage of the gear train is a sector gear which is connected to the ball valve shaft by means of a 1/4-inch diameter pin. The last stage sector gear and the ball valve shaft are concentric with respect to each other and are positioned within the actuator and guided by means of three radial bushings and two thrust bearings. On the oxidizer side of the bipropellant valve the pin that connects the sector gear to the top of the valve shaft is located in a slot in the sector gear such that during initial motion of the actuator the sector gear can rotate freely until the pin has completed its stroke within the slot. At that time the sector

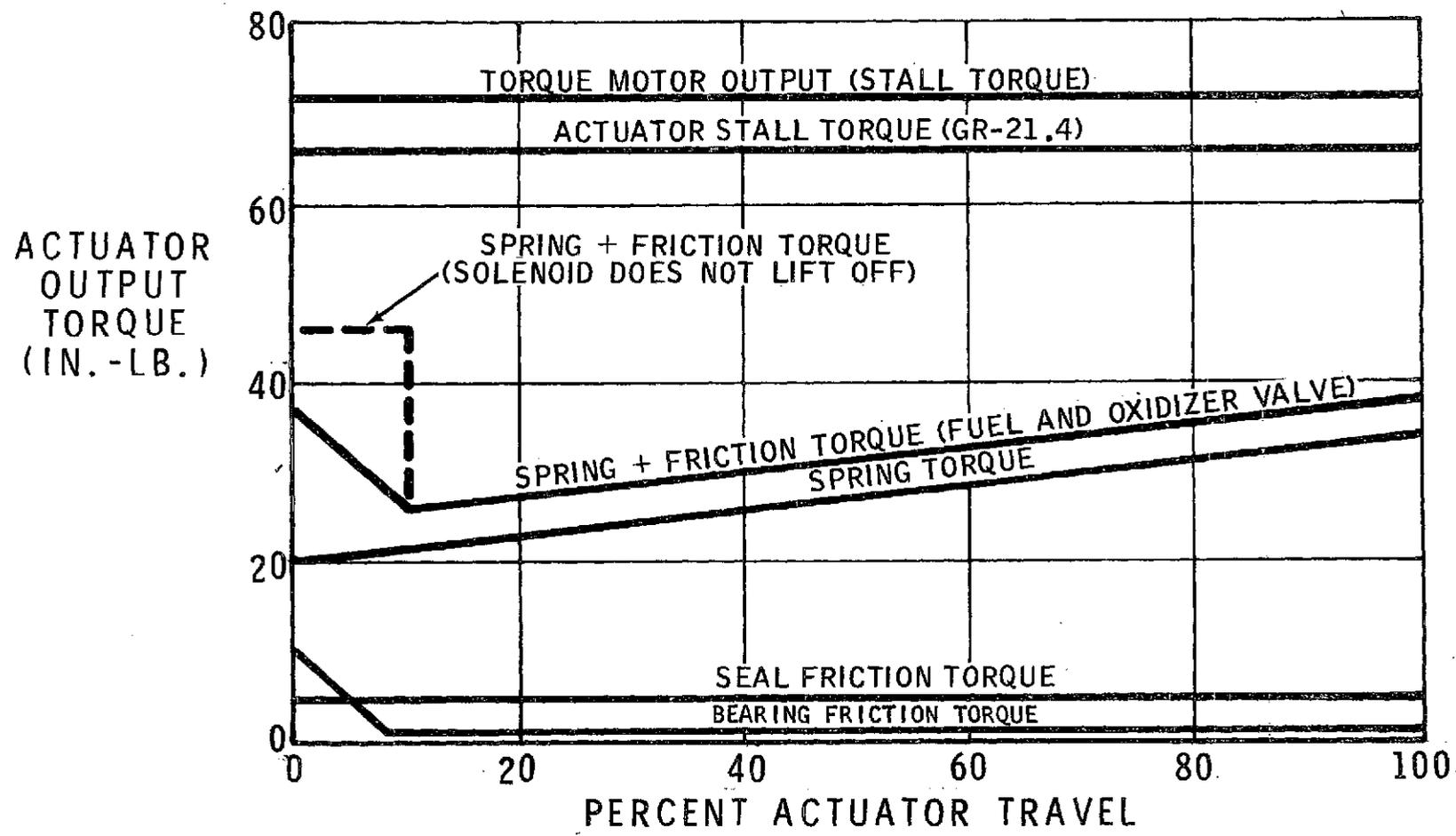
gear and the ball valve shaft rotate together. This slot and pin arrangement results in the delay of the start of motion of the oxidizer valve with respect to the fuel valve. Since the gear ratio to the oxidizer valve is higher than that to the fuel valve, the two valves nevertheless reach the open position at the same time.

The space between the valve shafts on the left and the right and between the gears on the top and the valve housing on the bottom is occupied by the electronic package. The package extends the entire length of the bipropellant valve. Access to the package may be gained through the cover plates either from the front of the valve or from the back of the valve. Wiring from the seat solenoids, potentiometers, and torque motor are all routed directly into the electronic package and are not exposed outside the valve housing, actuator housing or electronic package. All of the components of the actuation mechanism are made of teflon-impregnated 6061 aluminum with the exception of the electrical wiring, electronic components, magnetic materials in the torque motor, windings of the torque motor, the resistance element of the potentiometer, the potentiometer housing, and the torsional springs. The teflon-impregnated 6061 aluminum is utilized at load points throughout the actuator drive train such as at bearing surfaces and gears. This material has been proven in a number of space applications to be an excellent self-lubricating bearing surface.

5.2.3 Performance Characteristics

Utilizing the pressure drop and leakage analysis approaches discussed in Section 4.2.2, the ball valve sealing enclosure was sized and the actuation torques required to operate the valve were determined. Figure 5-11 shows the ball valve actuation torques as a function of the actuator travel. This figure shows the torque required due to bearing friction, seal friction, and the return spring as well as the sum of these three requirements. In addition, the dashed line in the figure represents the total torque required if the solenoid does not pull off the seal prior to the initiation of the ball rotation. As evident from Figure 5-11, an approximately 2 to 1 margin for actuator output torque to valve torque requirement has been provided in the design.

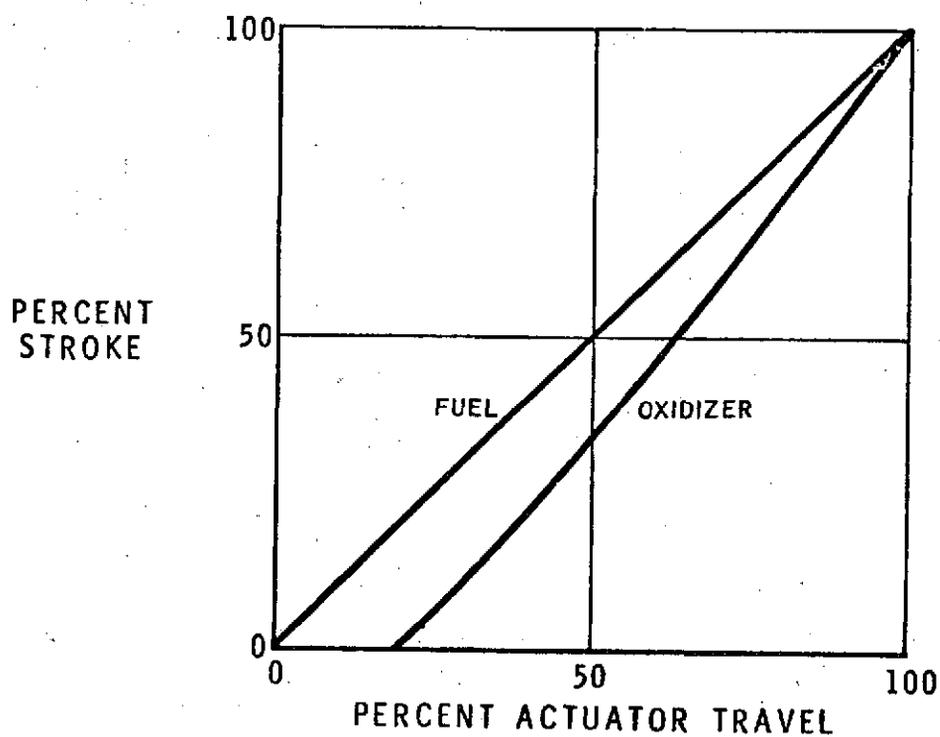
The same brushless DC torque motor was selected for the ball valve and for the dual poppet valve. This torque motor has already been described in Section 5.1. Also the characteristics of the electronic driver which is required for brushless DC torque motor operation were previously described in Section 4.2.2. The ball valve was also designed to feature a fuel lead similar to that of the poppet valve. The ball valve fuel and oxidizer rotation as a function of actuator travel are presented in Figure 5-12. The ball valve lead characteristics are somewhat different than those of the poppet valve because of the fact that the ball valve actuator simply employs different gear ratios to each of the valves whereas the poppet valve actuator employed a cam arrangement which was more flexible as far as designing for a specific fuel lead is concerned. The information presented in Figure 5-12 can be combined with that presented in Figure 4-15 to obtain the actual ball valve flow area as a function of time.



BALL VALVE ACTUATION TORQUES
 BI-PROPELLANT VALVE

5-22

Figure 5-11



LEAD-LAG CAPABILITY
BALL VALVE OPENING MOTION

Figure 5-12

The opening response of the ball valve is primarily a function of the velocity of the motor over the required rotational stroke. The average velocity of the motor is computed from the torque speed characteristics of the motor assuming an average load torque. Figure 5-6 shows the torque speed curve of the selected torque motor as a function of applied voltage and Figure 5-11 shows the torque characteristics of the motor and load. From Figure 5-11 the average load torque (spring plus friction torque) is approximately 20.5 inch-ounces at the motor (27 inch-pounds actuator output). Therefore, from Figure 5-6 the average motor speed is 62.5 radians per second. The total opening speed is then computed as the sum of the motor torque rise time ($T_m = 0.030$ sec.) and the average velocity.

The opening response characteristics of the ball valve were also determined as a function of torque motor electrical power. These data are presented in Figure 5-13. Thus in the event a faster opening response is desired or less electrical power becomes available the resultant impact on response or electrical power may be determined from this figure.

$$t_o = T_m + \frac{\Delta \theta_o (GR)}{\dot{\theta}_{M_{Avg.}}}$$

$$t_o = 0.57 \text{ sec.}$$

where

- T_m - Torque Motor Mechanical Time Constant (0.03 Sec.)
- $\Delta \theta_o$ - Valve Travel (90°)
- $\dot{\theta}_{M_{Avg.}}$ - Average Velocity Over Valve Stroke
- (GR) - Gear Ratio (21.4:1)

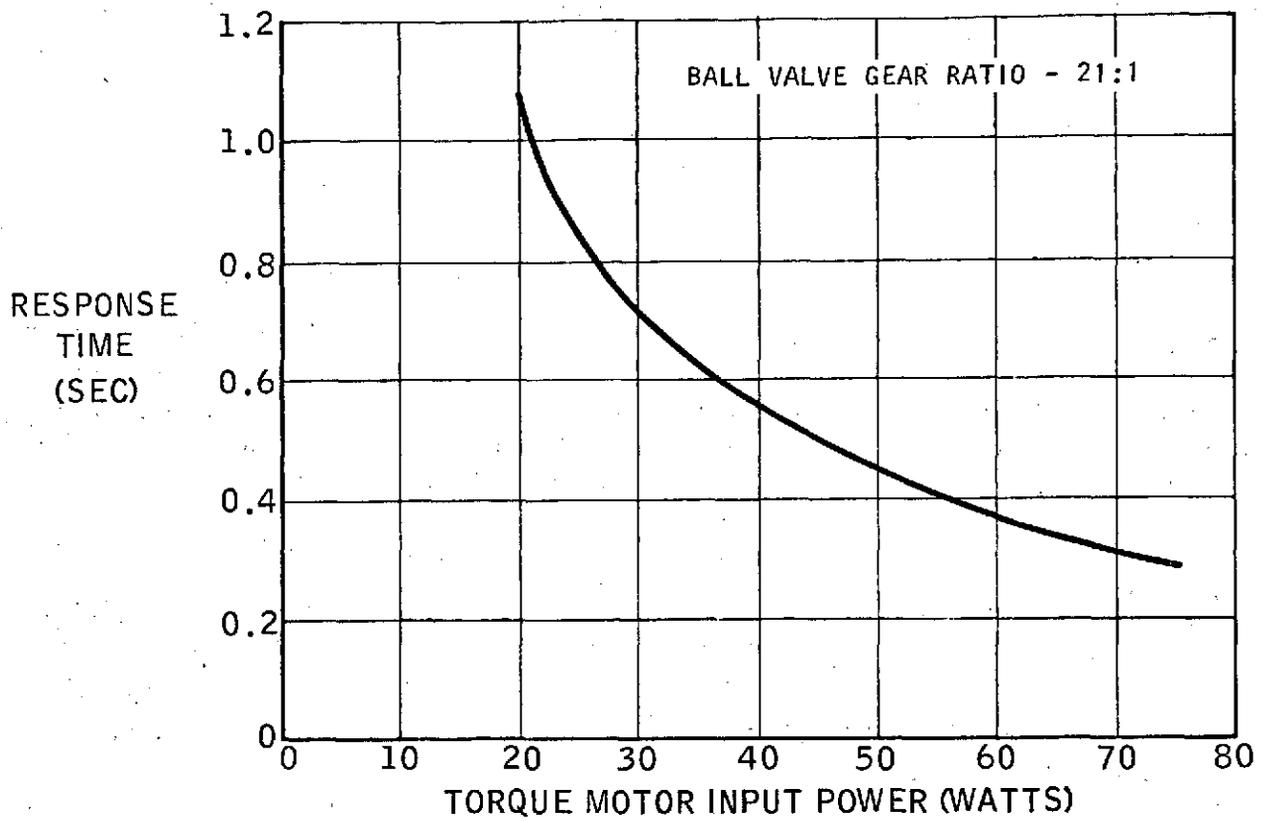
The closing response is a function of the net closing torque (spring torque minus friction torque) and the inertia of the complete valve. Dynamic breaking is also employed with the ball valve in a manner similar to that discussed in Section 5.1 for the poppet valve. The equation describing the total closing time is

$$t_c = \frac{I_m \dot{\theta}_{Max.}}{(T_s - T_f)} + \left[\Delta \theta - \frac{I_m \dot{\theta}_{Max.}^2}{(T_s - T_f)} \right] \frac{1}{\dot{\theta}_{Max.}}$$

$$t_c = 0.37 \text{ Sec.}$$

where

- $\Delta \theta$ - Valve Travel (90°)
- $\dot{\theta}_{Max.}$ - Maximum Closing Velocity of Motor at Impact (100 Rad./Sec.)
- I_m - Torque Motor Inertia (6.7×10^{-4} in. oz. Sec.²)
- $(T_s - T_f)$ - Closing Torque for Ball Valve



BALL VALVE OPENING RESPONSE

Figure 5-13

The electrical power requirements of the quadredundant ball valve are the same as far as the torque motor is concerned as those of the dual poppet valve. However, in addition a total of 120 watts per quadredundant valve is required to operate the solenoid actuators used to retract the ball valve seal. Thus the total peak power per quad valve is 312 watts. The solenoid electrical power is reduced to a holding power of only 40 watts per quadredundant valve once the valve has been opened consequently the total holding power required per quadredundant ball valve is 136 watts.

The sizing criteria used in the design of the solenoid actuator of the ball valve seal is presented in Table 5-IV.

TABLE 5-IV

SOLENOID DESIGN CRITERIA

- Force at 0.005 Inch Gap - 91 Lbs. at 9 Kilogauss
- Outside Diameter - 2.61 Inches
- Inside Diameter - 1.92 Inches
- Stroke - 0.006/0.005 Inch
- Coil Resistance - 29.4 Ohms
- Power - 15 Watts at 21 Volts

It should be noted that the 91 pounds of force specified is a conservative number assuming worst case tolerances for the seal effective diameter and the bellows effective diameter. The outside and inside diameter specified were arrived at on the basis of the ball valve design layout. The power specified pertains to one solenoid and there are two solenoids per bipropellant valve.

The ball valve design was stress analyzed in a manner similar to that of the poppet valve. Specifically, pressure vessel stress margins and stresses in the actuator drive train were the same as those for the poppet valve. A typical example of some of the stress analyses performed in support of the ball valve is presented in Appendix B. One of the more critical items requiring stress analysis for the ball valve is the single convolution bellows used as the compliant element in the seal design and the stress analysis for this bellows is also included in Appendix B.

Upon completion of the ball bipropellant valve design layout a detailed weight analysis was prepared. This analysis is summarized in Table 5-V. Based on this analysis a bipropellant valve weight of 7.12 pounds was determined. The corresponding quad valve weight is 28.48 pounds. This weight is almost 2 pounds heavier than the 26.6 pounds previously determined for the dual poppet valve. However, it should be noted that the pressure drop characteristics of the ball valve are only 1.0 psi compared to 1.4 psi for the poppet valve. Both of the valve concepts offer very attractive low weight characteristics.

TABLE 5-V

BALL VALVE WEIGHT

VALVE COMPONENTS	WEIGHT - POUNDS			ACTUATOR COMPONENTS	WEIGHT LBS.
	Al	Ti	Cres		
COIL WIRE (CU.)			0.116	TORQUE MOTOR	0.610
SOLENOID RING (In. 600)			0.036	TORQUE MOTOR COVER	0.176
SOLENOID FLANGE (In. 600)			0.174	ROTOR SHAFT	0.051
SOLENOID OUTER RING (In. 600)			0.261	MAIN COVER GEAR HOUSING	0.278
SOLENOID FIELD (446)			0.465	BASE FRAME	0.121
SOLENOID ARMATURE (446)			0.148	DRIVE GEAR (2)	0.087
BELLVILLE (In. 718)			0.088	SECTOR GEAR (2)	0.065
SEAT RETAINER (In. 718)			0.084	BALL DRIVE SHAFT (2)	0.119
BELLOWS (SEAT) (In. 718)			0.004	SPRING (2)	0.141
BUSHINGS	0.026		(0.078)	BUSHING (12)	0.054
BALL VALVE ASSEMBLY (455)			0.271	ELECTRONIC COVER PLATE	0.073
BALL VALVE SUPPORT	0.102	(0.166)		POT (2)	0.074
SEAL RING (In. 600)			0.048	ELECTRONICS	0.250
VALVE HOUSING	0.478	(0.781)			

NOTE: Brackets indicate Oxidizer Valve.

- FUEL VALVE ————— 2.30 LBS.
- OXIDIZER VALVE ————— 2.72 LBS.
- ACTUATOR ————— 2.10 LBS.
- TOTAL BI-PROPELLANT VALVE — 7.12 LBS.
- QUAD PACKAGE WEIGHT ————— 28.48 LBS.

The performance characteristics of the quad redundant ball valve are listed in Table 5-VI. These characteristics are in accordance with the performance requirements of

TABLE 5-VI
PERFORMANCE CHARACTERISTICS - QUAD REDUNDANT BALL VALVE P/N L14091

Pressure Drop	- 1.0 psi @ 11.9 lbs/sec N ₂ O ₄ (for inlet & outlet manifolding add 0.25 psi)
Operating Pressure	- 250 psi nominal - 500 psi proof
Opening Response	- 570 ms @ 250 psi inlet pressure, 24-30 volts DC, 40-200°F
Closing Response	- 340 ms @ 250 psi inlet pressure, 11.9 lbs/sec N ₂ O ₄ volts DC, 40-200°F
Internal Leakage	- 10 SCC/Hr Helium maximum @ 2-250 psi inlet pressure
External Leakage	- 1.66 x 10 ⁻⁷ SCC/sec Helium maximum per joint @ 2-250 psi inlet pressure
Operating Voltage	- 24-32 vdc
Electrical Power	- 312 watts to open - 136 watts to hold open after 1 sec
Operating Temperature	- 40-200°F
Cycle Life	- 4000 wet, 6000 dry
Service Life, Maintenance Free	- 500 missions or 10 years
Weight	- 28.5 lbs.
Acceleration	- 4 g's maximum
Compatibility	- N ₂ O ₄ and MMH in combination with their combustion products, moisture, and salt spray

Section 3 and reflect the analysis data presented in the preceding paragraphs. Particular emphasis has been placed throughout the design effort in obtaining the cycle life, service life, and reuseability characteristics required for the Space Shuttle. High cycle life of the sealing closure is achieved through the use of low sealing stresses, the elimination of scrubbing at the sealing closure interface, and a plastic/metal materials combination. Potential wear points within the valve propellant cavity have been minimized and consist of only the bushings required to support the ball and the shaft seals. The stops required to limit the opening motion and the closing motion incorporate spring elements which have been sized to absorb the kinetic energy of the moving mechanism. The materials of construction utilized in the bushings of the valve have been previously employed in similar ball valve designs. All materials employed feature the best possible compatibility ratings with both of the propellants as well as with their combustion products in combination with moisture and salt spray. The wear points in the actuator have been carefully analyzed to feature controlled operating stresses and are lubricated by means of teflon impregnated aluminum to assure high cycle life.

THIS PAGE LEFT INTENTIONALLY BLANK

6.0 FABRICATION

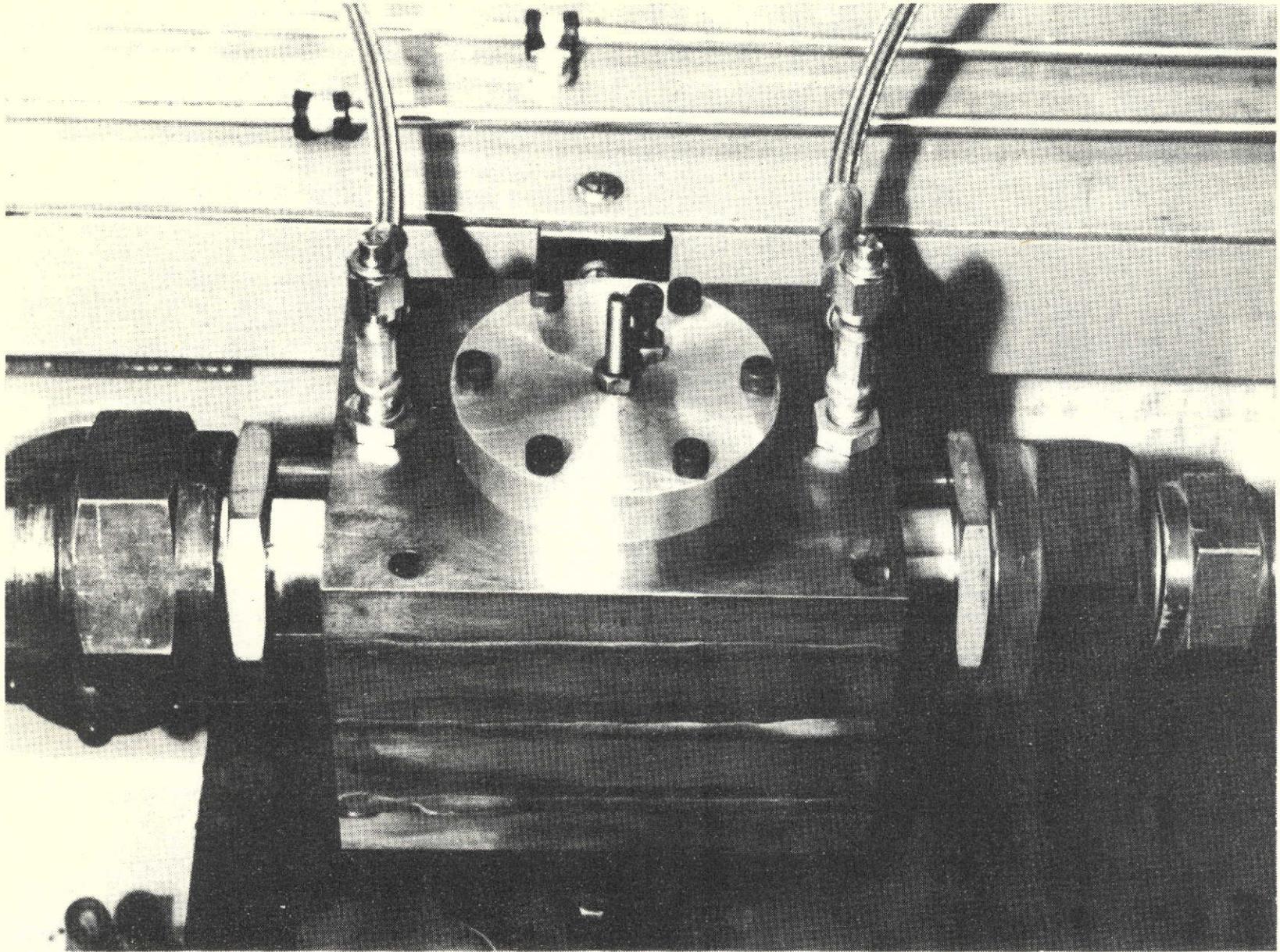
Two major components were fabricated during this program. These are the flow fixture simulating the flow passage through a dual poppet valve and the complete bi-propellant ball valve including actuation system. In addition, several minor fixtures required during the fabrication and testing of the ball valve were also made. These included valve inlet and outlet adapters, seal installation tools, and an electric motor torque measuring tool. All fabrication was accomplished in accordance with Marquardt's experimental hardware manufacturing procedures. During Marquardt's experimental hardware program, a liaison engineer is assigned the responsibility of directing all detail parts manufacture and component assembly. The liaison engineer initially prepares a fabrication schedule and designates which of the parts are to be built in-house in the experimental shop and which are to be subcontracted. Through the purchasing department, he coordinates the purchase of all parts required during component assembly. The liaison engineer maintains a log book for each component which contains a drawing of the detailed parts as well as copies of all purchasing and shop directives. If during the manufacturing process parts are fabricated which do not fully meet the drawing requirements, the liaison engineer and the project engineer review the discrepancy and disposition of the part. If the part is accepted as is, the drawing in the log book for that component is updated to reflect the as built condition. In this manner, a thorough record of the as built condition of the component is kept. The liaison engineer actually participates in the final assembly of the component, as does the development engineer who will subsequently be responsible for the test evaluation of the component. Fabrication techniques of the parts manufactured during this program are briefly described in the following two sections.

6.1 FABRICATION OF THE DUAL POPPET VALVE FLOW FIXTURE

The dual poppet valve flow fixture was previously shown in Figure 5-14. A photograph of the flow fixture is shown in Figure 6-1. As shown in the photograph, flow through the fixture was from left to right. The two 1/4" fittings on top of the fixture are for measurement of the pressure drop across the flow fixture. Also evident from this picture is the set screw atop the flow fixture which is used for adjustment of the poppet stroke.

The flow fixture body was made entirely of 300 series stainless steel and the poppet and cover plate were made from aluminum. To facilitate contouring of the flow passage as it enters and leaves the poppet/seat interfaces, the flow fixture was designed to be made from four plate sections. These sections were then subsequently brazed together to complete the flow fixture body. The braze joints of the flow fixture body are evident from Figure 6-1. All parts internal to the flow fixture body were fabricated separately and were installed through the top and through the bottom of the flow fixture. The fixture employed one static seat at each of the seats, one static seal at the lower cover plate, and two static seals at the upper cover plate.

NEG. 73-337-1



6-2

Figure 6-1

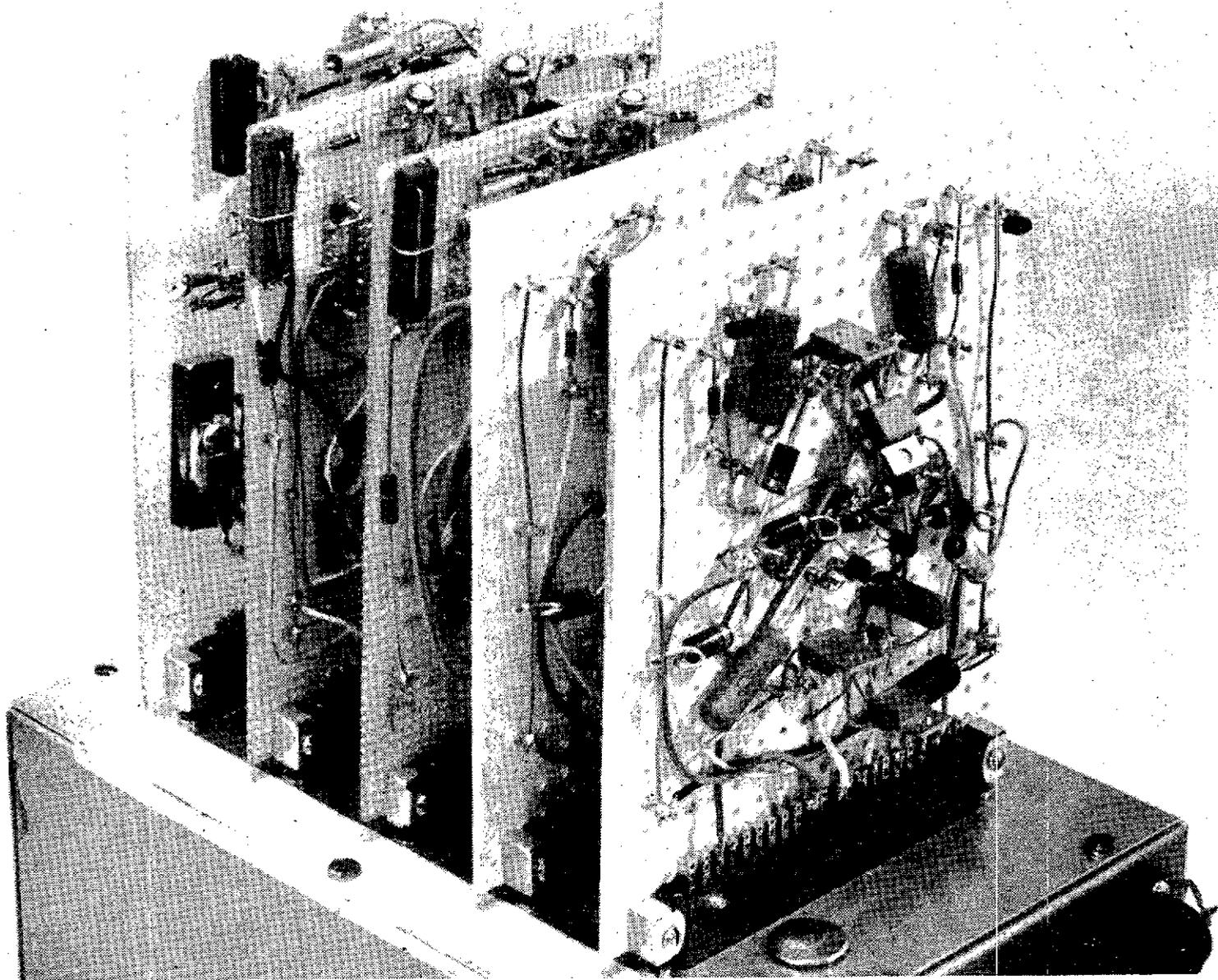
DUAL POPPET VALVE FLOW FIXTURE

The dual poppet valve flow fixture was fabricated entirely in the Marquardt experimental shop except that the brazing of the four plates to complete the body was performed by a subcontractor. Fabrication of the flow fixture body proved to be substantially more difficult than had originally been anticipated. The principal problem was the fact that the flow passage contour consisted of slots requiring a number of different radii and tapers and that the machining of these slots as well as the verification of the correct dimensions was extremely time consuming. After the internal flow passage dimensions were machined into the four plates the four plates were brazed together in a furnace. This brazing cycle had to be repeated three times by the subcontractor until a leak proof body was achieved. After brazing of the assembly the inlet and outlet fitting holes were machined and the fittings were welded to the brazed body. Leakage problems with the fitting to body weld joint were also encountered and this joint had to be repaired three times. Fabrication of the flow fixture internal parts progressed very smoothly as did the final assembly of the flow fixture. The flow fixture was subsequently tested in the building 37 flow facility as described in Section 7.1.

6.2 BIPROPELLANT BALL VALVE AND ACTUATION SYSTEM FABRICATION

The valve and actuation system fabricated during this program consisted of two distinct components. One is the ball valve including electrical torque motor and the other is the breadboard electronic driver required to operate the torque motor. A photograph of the breadboard electronic driver is shown in Figure 6-2. The electronic driver was assembled entirely in the Marquardt electronic laboratory using a standard chassis and standard circuit boards. All electronic components were of the same type as specified for the flight weight driver except that commercial rated components were considered acceptable. In addition to those components normally employed in the flight weight driver the driver also included a number of variable resistors and other components which permitted the adjustment capability previously described in Section 5 of this report. Six of these variable resistors are visible at the top of the first three cards from the left in the Figure 6-2. The five cards visible in Figure 6-2 represent the following portions of the electronic circuit. The card on the left is for the dynamic breaking required during the closing motion of the valve and also includes a voltage regulator. The second and third cards from the left are the control logic for each of the two motor windings. The fourth and fifth cards from the left are the drivers for the two motor windings and include the heaviest components of the electronic circuit namely the power transistors. Fabrication of the entire electronic driver was fairly routine and was accomplished in a period of two weeks after the components had been obtained.

The actuation system of the torque motor operated ball valve is an integral part of the ball valve with the exception of the breadboard electronic driver. In the final flight weight version this driver will also be integrated into the ball valve housing. The assembly drawing for the prototype ball valve and actuation system as it was built during this program is shown in Figure 6-3. A detailed parts list identifying all valve components as well as the number of parts required is presented in Appendix C. Fabrication of the



BREADBOARD ELECTRONIC DRIVER

6-4

Figure 6-2

Page intentionally left blank

prototype ball valve was initiated in December 1973 and was originally scheduled to be completed late in March 1974. A breakdown of the fabrication schedule showing all machined parts and major subassemblies is presented in Figure 6-4. The pacing item in the fabrication schedule was the receipt of the part number X29643 flexure from the bellows vendor and its subsequent incorporation into the solenoid assembly. This flexure consists of a single convolution bellows with special end fittings. As discussed in a subsequent paragraph inability of the supplier to fabricate the single convolution bellows greatly extended the valve fabrication time from that shown in Figure 6-4.

When the fabrication of the bipropellant ball valve was initiated a review of the work load in the Marquardt experimental shop was made to determine if the planned schedule could be accomplished. This review indicated that the experimental shop was heavily overloaded and it was therefore decided to subcontract nearly all detail parts and to limit the work done by Marquardt's experimental shop to possible rework and fit up during assembly. Consequently bids were solicited from a number of subcontractors for fabrication of the detailed parts and at the same time the material required for this fabrication was procured by Marquardt for subsequent forwarding to the winning bidder.

The bipropellant ball valve includes a number of purchased parts. These are such parts as screws and bolts used during the assembly of the valve; omni seals used at the shafts, shaft seal leak check ports, and solenoid assembly; position indicators; electrical connector; and the brushless DC torque motor including Hall effect devices. Delivery times of these various purchased parts were compatible with the overall fabrication schedule except for the brushless DC torque motor. The brushless DC torque motor featuring a 55 in/oz. stall torque requested by the Marquardt Company constituted a slightly larger version of another torque motor which featured a 45 in/oz. stall torque and which was on the shelf at the torque motor supplier. To obtain the Marquardt designated torque motor would have required a delivery time of 6 months due to a heavy work load at the torque motor supplier. Since this was incompatible with the overall fabrication schedule it was decided to use the off-the-shelf motor with the slightly lower operating torque. This motor was readily available and required only minor modification to the actuator housing (installation of a spacer) to accommodate it. The only disadvantage in using this motor was the fact that a lower operating margin was available (approximately 60% as opposed to the originally intended 100%) and that the valve opening response at a particular voltage would be somewhat slower. However the same response could still be achieved by simply running a slightly higher voltage.

Fabrication of the detailed parts at the outside vendors proceeded in a reasonable manner. Minor problems included the fact that the vendor fabricating the balls for the bipropellant valve located the shafts inappropriately with respect to the ball sealing surface and therefore had to remake the balls, the supplier manufacturing the gears did not properly allow for the nituff coating which increased the gear finished dimension such that the gear location in the gear box had to be modified to prevent interference between the gears, and

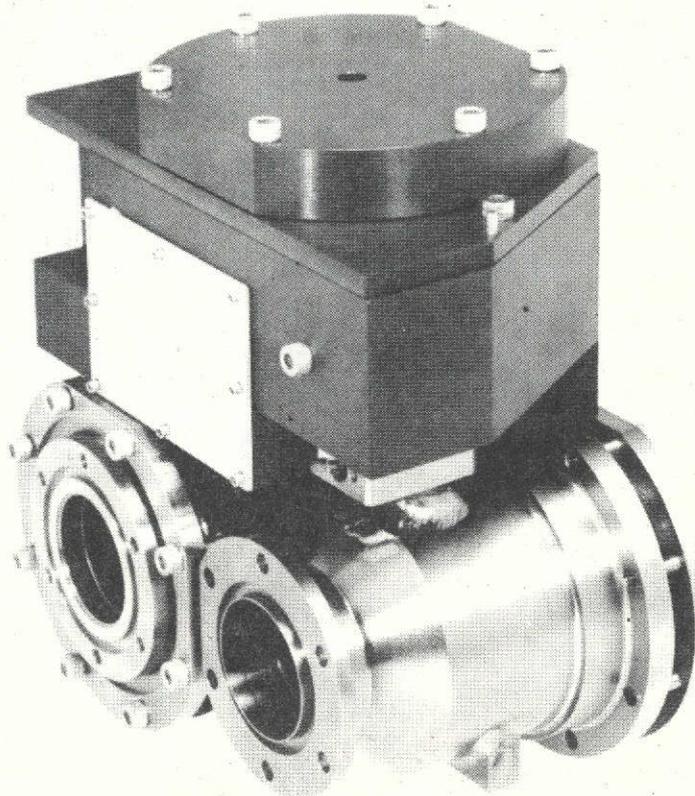
some other minor dimensional problems which did not significantly impact the valve design. However, the one major problem that occurred during the manufacture of the bipropellant valve was the inability of the bellows vendor to fabricate the flexure assembly (which contains a single convolution bellows) capable of meeting the effective bellows area requirement within the specified tolerance of plus or minus 1%. This test was to be witnessed by Marquardt liaison engineering since it is a critical requirement in the operation of the valve because it determines the load between the seal and the ball and therefore effects leakage and seal retraction characteristics.

During the witnessing of the bellows effective area tests it became apparent that the bellows vendor had not recently performed this type of test and did not have an understanding of the accuracy of the test set-up required to demonstrate a plus or minus 1% effective area accuracy. Furthermore the vendor did not appear to have sufficient control in his welding set-up to be able to control the bellows effective diameter sufficiently to meet the plus or minus 1% effective area requirement. The liaison engineer intermittently spent a number of days with the vendor while the vendor was improving his test set up and while he made attempts to demonstrate the effective area of this bellows. However, after several weeks had passed and the data obtained at the vendor still indicated an average effective area outside of the tolerance band as well as a data scatter greater than the tolerance band project engineering decided to initiate an in-house effort to machine the single convolution bellows from bar stock. While it was difficult to machine the particular bellows configuration the Marquardt Company was able to successfully fabricate the required two single convolution bellows and to demonstrate the effective area of these bellows in a period of approximately two months.

The Marquardt machined flexure assemblies (including the single convolution bellows) were subsequently forwarded to another supplier for installation into the solenoid assembly. Unfortunately during the installation process this supplier accidentally damaged one of the single convolution bellows. The same supplier then machined a new bellows and submitted it to Marquardt for test verification. After two submittals this bellows was considered acceptable and the solenoid assemblies were subsequently completed. However, the necessity for making a new bellows resulted in another major program delay.

During final assembly of the bipropellant ball valve at the Marquardt Company several other minor problems arose. One of these problems was the discovery that some interference existed between the ball flow tube and the inner liner as well as the seal as the ball rotated from the closed to the open position. Consequently the flow tubes had to be trimmed slightly. Two other minor problems were discovered in the actuator where insufficient clearance was observed between the torsion spring inner diameters and the adjacent cylindrical portion of the gear case and it was also determined that the slot in the sector gear of the oxidizer valve had been located improperly. There were several other minor problems which were corrected in the experimental shop as soon as they were observed.

The completed bipropellant ball valve assembly is shown in Figure 6-5. The valve located in the foreground of the picture is the oxidizer valve with the inlet on the left side and the outlet on the right side. The fuel valve is adjacent to this valve and its outlet is shown on the left side of the picture. Above the two valves is the actuator gear box and torque motor housing. The black appearance of these parts results from the Nituff treatment of the aluminum. The bright rectangular shaped cover on the gear box is a cover on the cavity that is to contain the flightweight electronic driver of the valve. This cavity is slightly smaller in cross section than the cover and extends the length of the gear box. A similar cover is installed on the opposite side of the gear box. The circular shaped housing with the two flats located atop the valve contains the pancake brushless DC torque motor and Hall effect devices.



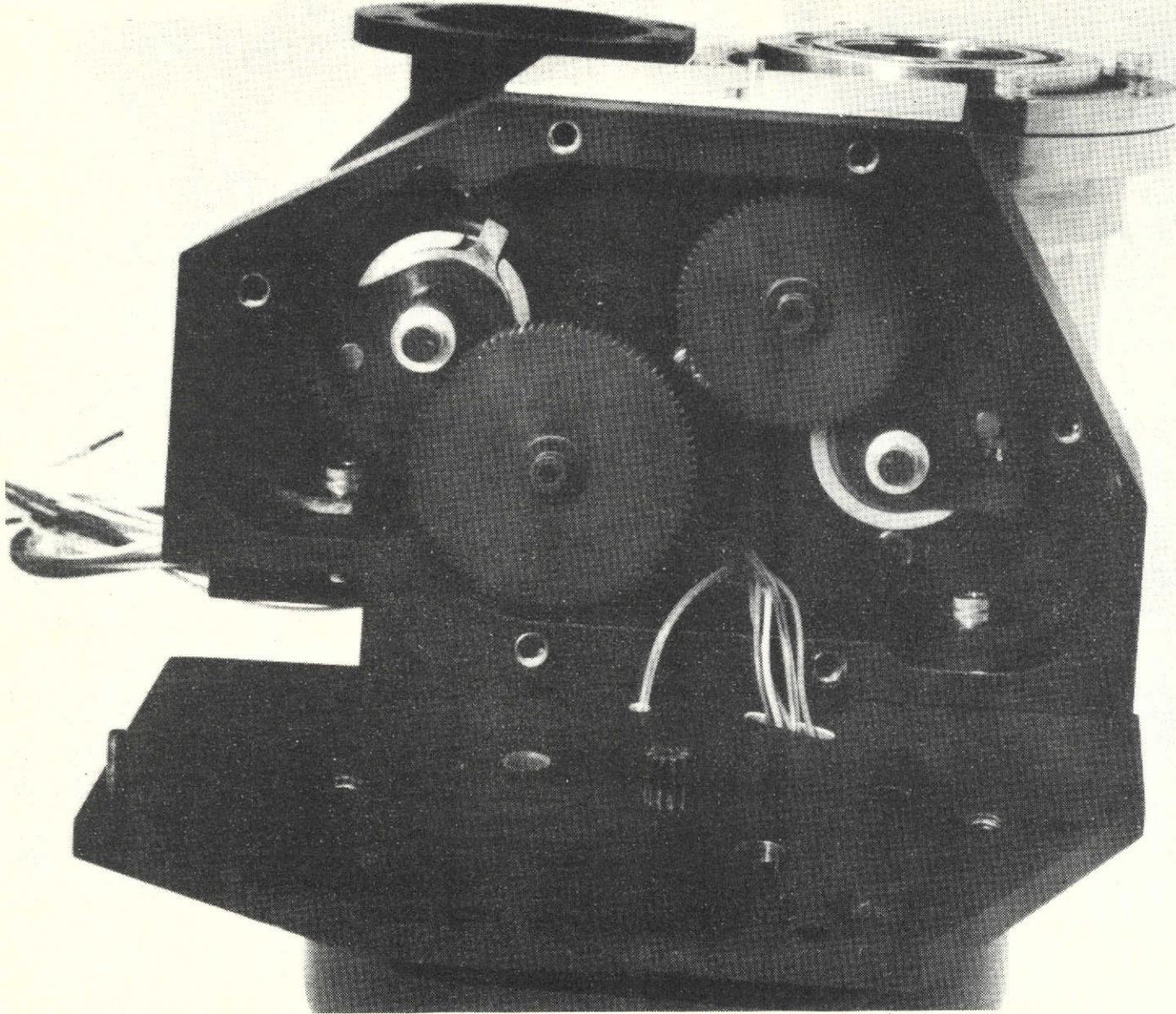
NEG. 74-348-30

BIPROPELLANT BALL VALVE ASSEMBLY

Figure 6-5

A view into the actuator gear box is shown in Figure 6-6. In this view, the oxidizer valve is on the right side and the fuel valve on the left side. The small driving gear from the torque motor may be seen protruding through the tilted up cover plate. The wiring to the torque motor is routed through the cover plate and through the gear box into the electronic driver cavity. The torque motor output gear simultaneously drives the two large gears seen

NEG. 74-356-10



0 Inch 1

ACTUATOR GEAR BOX

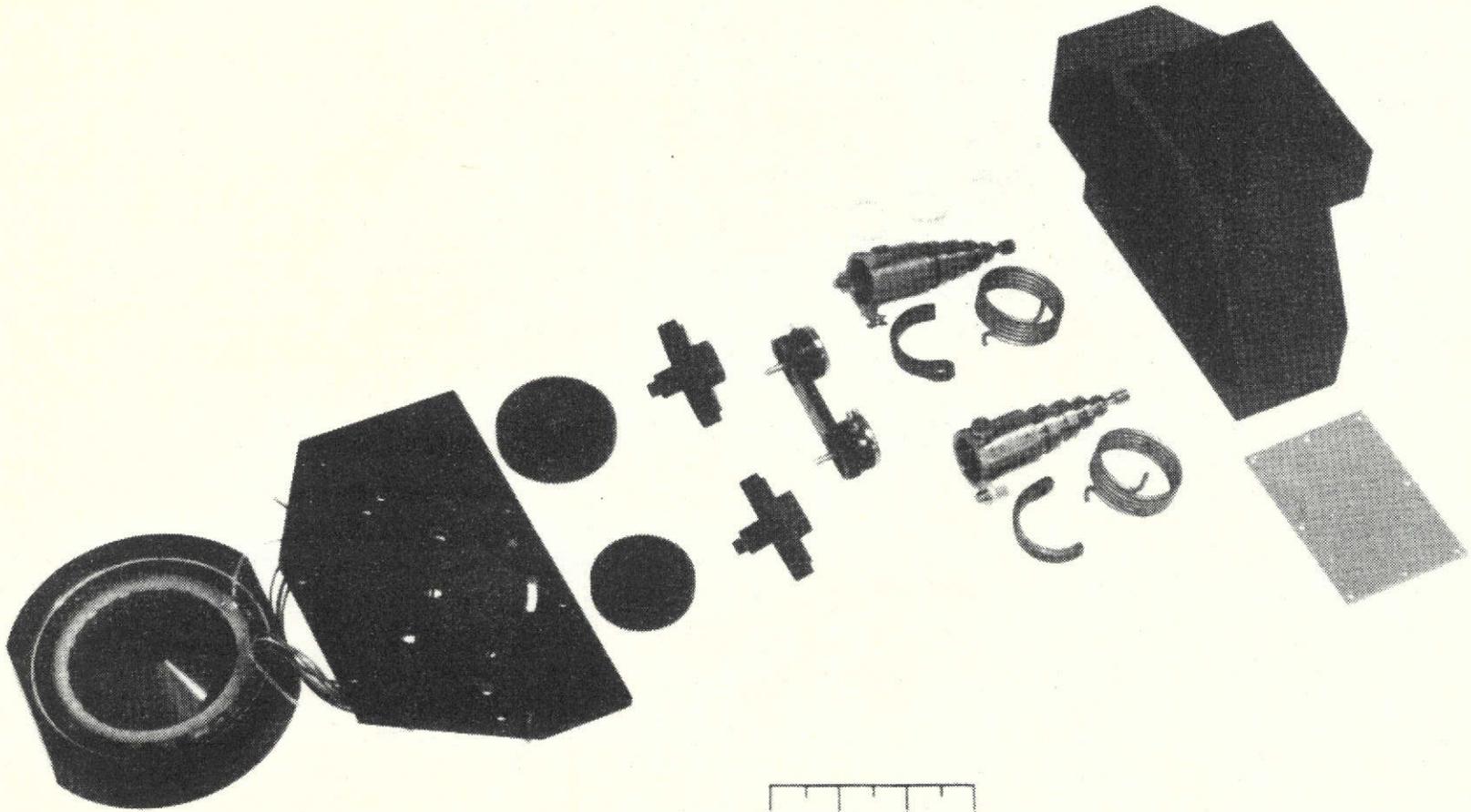
6-10

Figure 6-6

near the center of the gear box. These gears, in turn, through reduction gears, drive the sector gears located at each of the valve shafts. A slot and pin arrangement in the oxidizer valve sector gear results in that valve remaining closed until the gear has turned such that the pin has traversed the slot. Consequently, a fuel lead is obtained. Also visible in this photograph are the two C shaped impact springs. The one located near the oxidizer valve constitutes the stop for the closed position, and the one located near the fuel valve constitutes the stop for the open position.

An exploded view of the actuator is shown in Figure 6-7. From left to right, the following components are evident: brushless DC torque motor including output gear, cover plate, two main drive gears, two sector gears, two position indicators, two valve shafts, two impact springs, two torsion return springs, eight shaft seals, the gear box, and the electronic driver cavity cover plate.

The valve components, except for the solenoid assembly and solenoid retainer rings, are shown in Figure 6-8. The assembled fuel valve, except for the solenoid assembly and retainer ring, is shown on the right; and an exploded view of the components of the oxidizer valve, except for the solenoid assembly and retainer ring, is shown on the left. The scalloped ball and the shaft bearings are apparent on the left side of the exploded view. Between the scalloped ball and the oxidizer valve housing the valve liner may be seen. This liner serves two purposes. It permits the alignment of ball and seal external to the valve housing and it also routes the purge gas from the inlet of the valve to the relatively dead volumes adjacent to the single convolution bellows near the outlet of the valve. A photograph of the solenoid assembly which includes the solenoid actuator and the seal is shown in Figure 6-9. The metal-backed teflon seal is readily visible at the top of this picture as are several holes in the solenoid structure which constitute the purge passage to the volume around the outside of the single convolution bellows. The gap in the outer cylindrical portion of the solenoid assembly near these purge holes is the stroke of the solenoid through which the seal is retracted. The wiring shown near the bottom of the picture is only temporarily placed there. Normally it is routed radially away from the solenoid assembly directly into the electronic driver cavity.



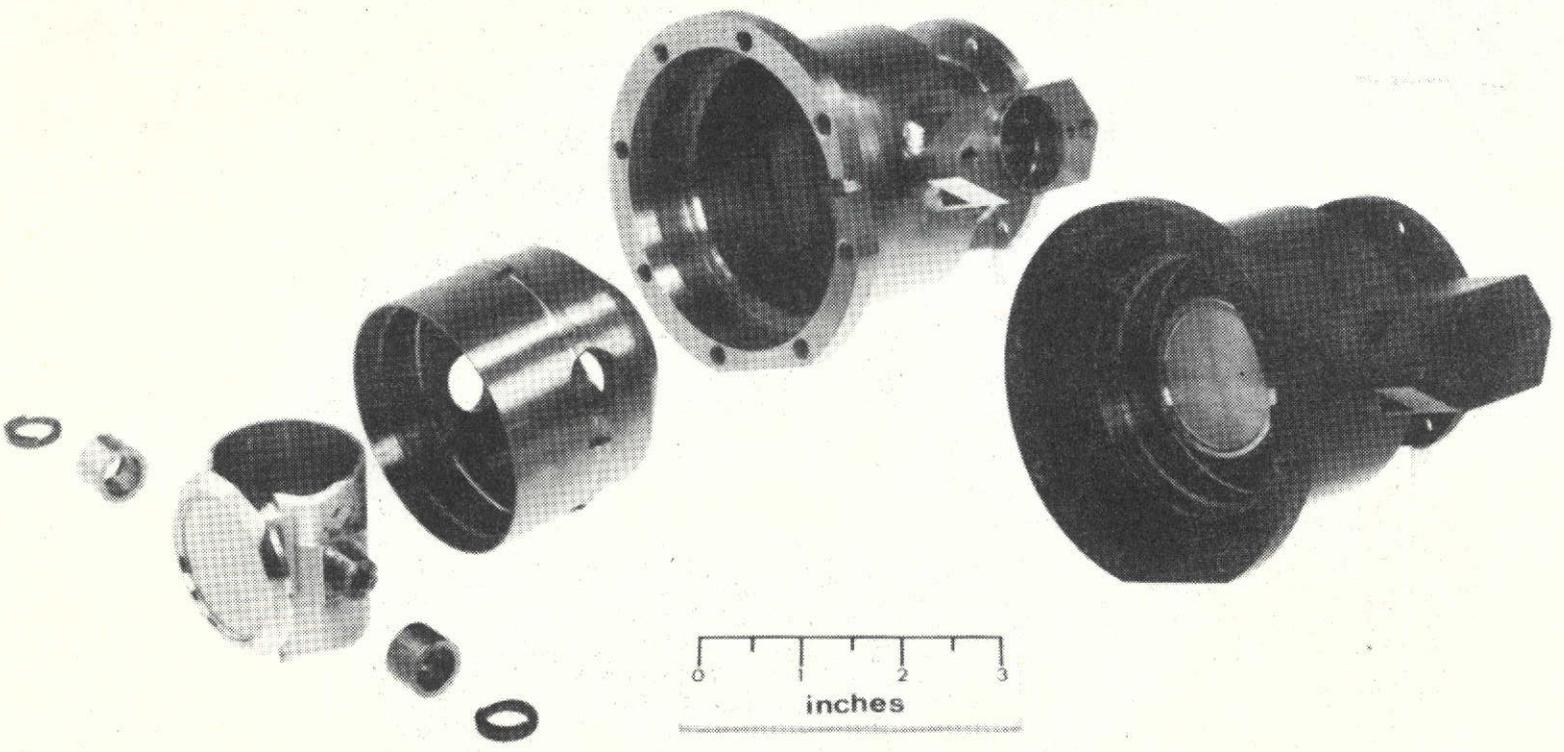
0 1 2 3
inches

ACTUATOR - EXPLODED VIEW

NEG. 74-3556-3

Figure 6-7

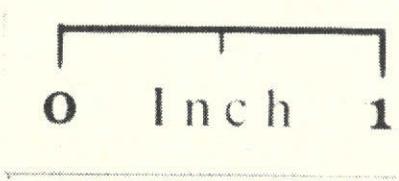
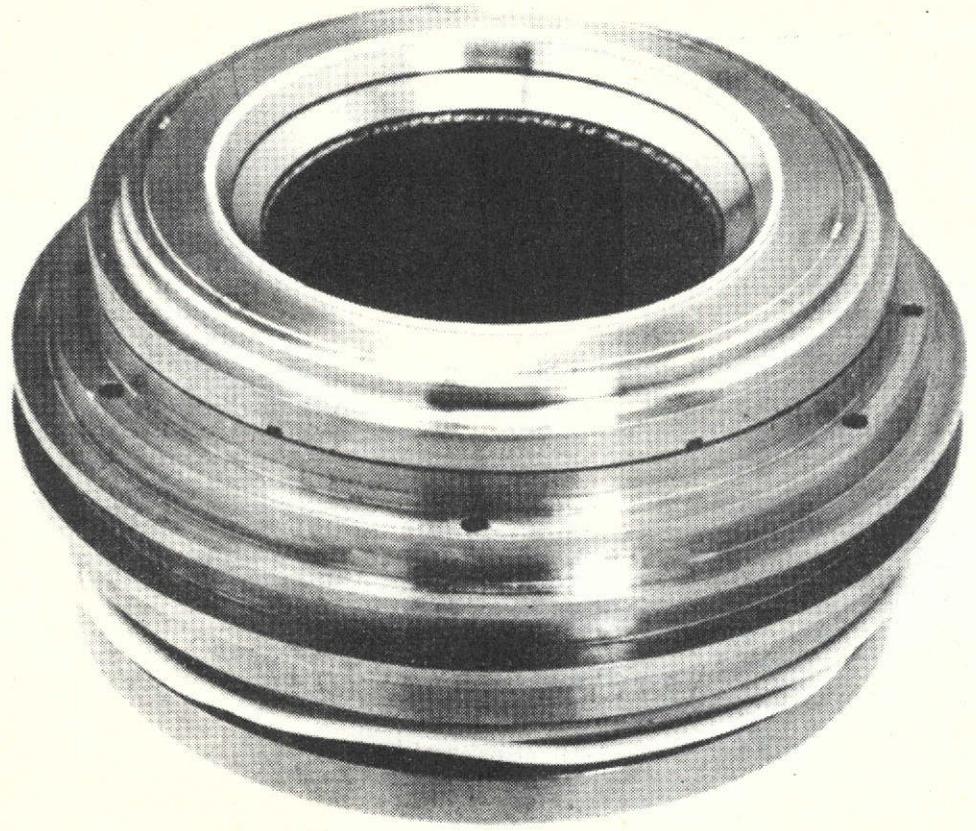
6-12



VALVE COMPONENTS

NEG. 74-356-2

Figure 6-8



SOLENOID ASSEMBLY

6-14

Figure 6-9

7.0 TESTING

The developmental testing performed during this program consisted of the evaluation of two major components. These are the flow fixture simulating a dual poppet valve and the bipropellant ball valve and actuation system. The resultant test data are presented in the following sections.

7.1 DUAL POPPET VALVE FLOW FIXTURE TESTING

The purpose of this test series was the determination of the pressure drop characteristics of the test fixture. The test fixture was installed in the water flow system in Building 37 at the Marquardt Van Nuys Test Facility. A photograph of the test set-up is shown in Figure 7-1. The flow fixture may be seen at the bottom of this photograph. Flow is from right to left. A turbine flow meter was used to measure water flow rate and the delta p gauge located in the center of the picture with a range of plus or minus 1.0 psi was used to monitor the pressure drop through the flow fixture.

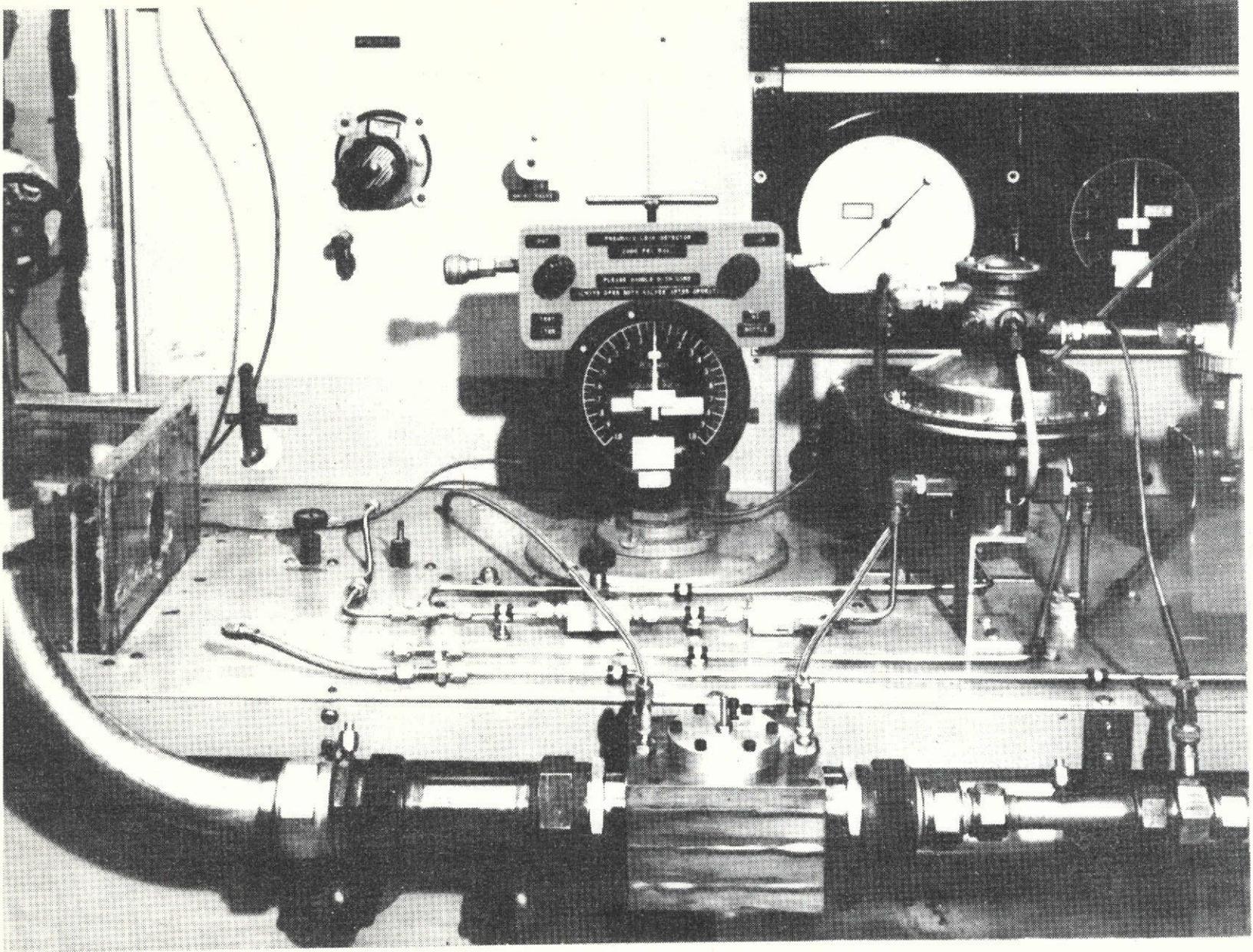
The test results are shown in Figures 7-2 and 7-3. Figure 7-2 compares the actual pressure drop with the predicted pressure drop when the valve is in the wide open position. As can be seen in this figure the measured pressure drop compares very closely with the predicted pressure drop which was based almost entirely on orifice flow through the metering edges of the poppet/seat. It is concluded from this factor that a combination of judicious design of the valve inlet and outlet transition passages along with keeping velocities down to a minimum level was successful in eliminating most of the pressure drop due to turning despite the rather complex flow path. Figure 7-3 shows the pressure drop at intermediate valve openings and confirms the fact that the valve can be considered a simple orifice following normal orifice flow equations through its stroke.

7.2 BIPROPELLANT BALL VALVE AND ACTUATION SYSTEM TESTS

The bipropellant ball valve and actuation system consists of the torque motor operated bipropellant valve and the breadboard electronic driver described previously in Section 6. The test program consisted of a series of component tests involving only the torque motor and electronic driver and subsequently of the entire valve and actuation system. Testing was performed in accordance with Marquardt Test Plan MTP 0220 which is presented in Appendix D.

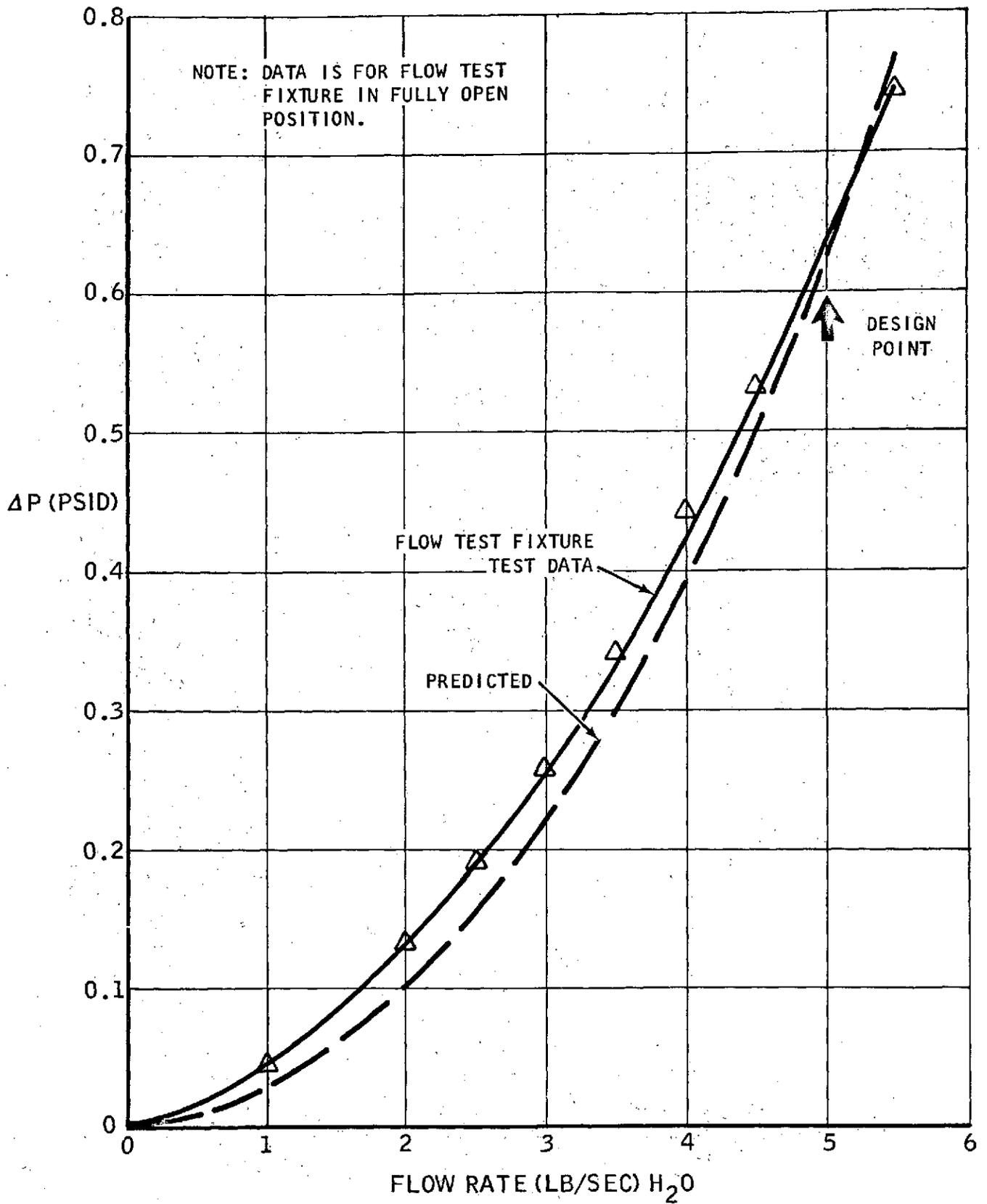
7.2.1 Torque Motor/Driver Testing

The purpose of these tests was to integrate the brushless DC torque motor with the breadboard electronic driver and to determine the no load and stall torque characteristics of the torque motor. These tests were conducted in the electronics laboratory (Building 32) at the Marquardt Company.



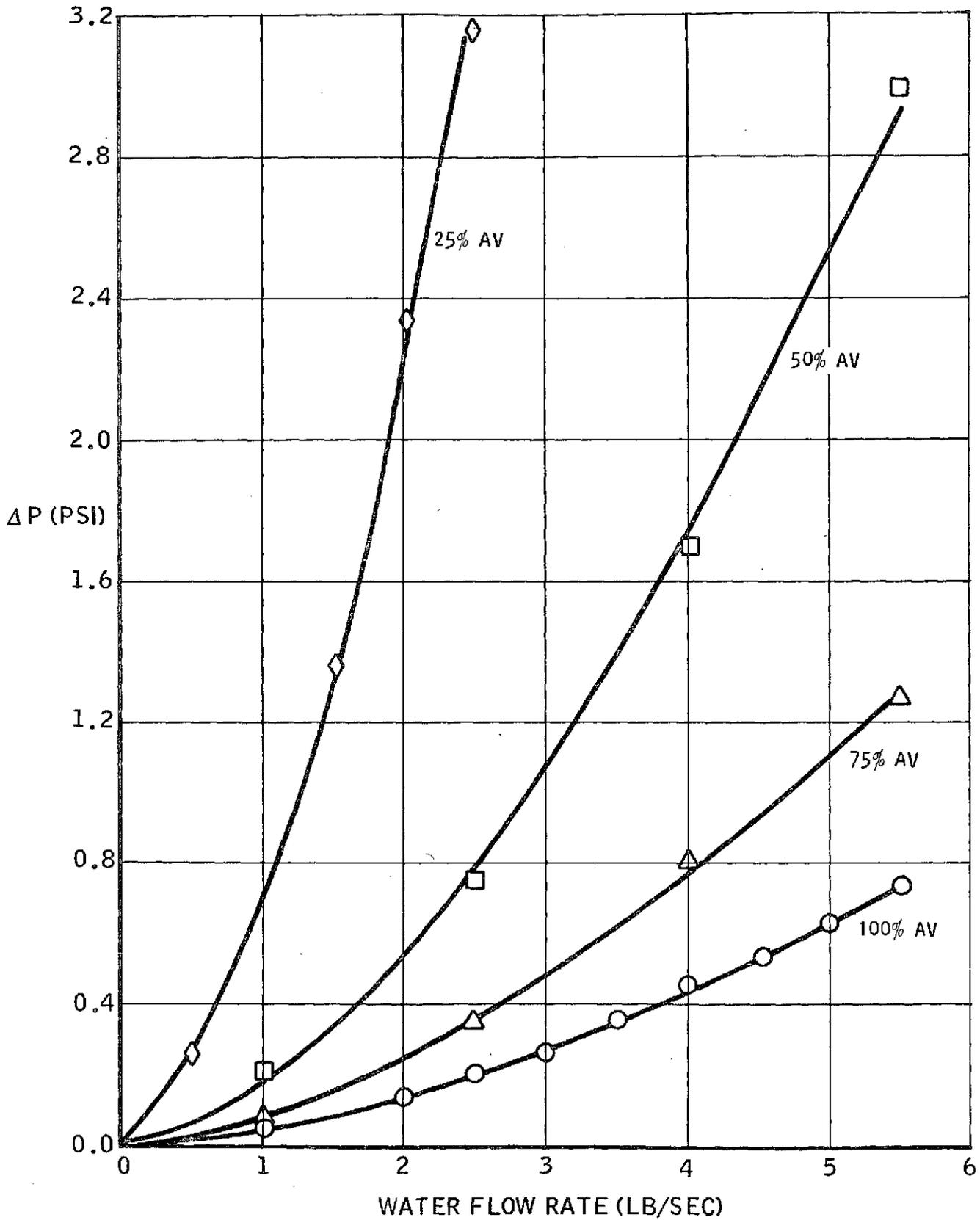
POPPET VALVE FLOW FIXTURE TEST SET-UP

Figure 7-1



DUAL POPPET VALVE PRESSURE DROP VS WATER FLOW RATE

Figure 7-2



DUAL POPPET VALVE PRESSURE DROP VS WATER FLOW RATE AND OPEN POSITION

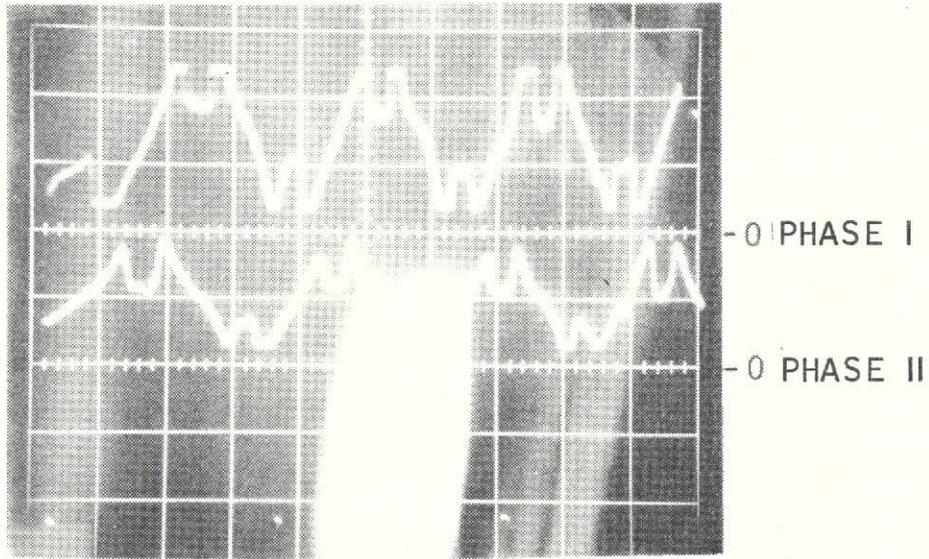
For the component tests the electric torque motor and the driver were located on a work bench. Torque motor speed was measured with a stroboscope and motor output torque was measured with a torque wrench which was attached directly to the motor shaft. A dual beam oscilloscope was used to monitor voltage and current signals at various points in the electronic driver.

As previously shown in Figure 4-17 it was the intent to set up the electronic driver to supply the torque motor with a 60% duty cycle. During the trimming of the driver circuits to obtain this duty cycle it became apparent that the hall effect device output signals were not of a pure sawtooth shape but rather were as shown in Figure 7-4. It may be seen from this oscilloscope photograph that the output signals include an additional dip at the peak of the sawtooth pattern. This dip is due to the finite spacing of the permanent magnets in the rotor. Since the driver circuit was designed to trigger at an adjustable rising voltage signal to start a pulse and to terminate this pulse at an adjustable decreasing voltage signal it became apparent that the level of this triggering voltage had to be set lower than the bottoms of the dips from the magnet spacing. Thus to prevent the erroneous triggering of the driving signals during the magnet's dips it became necessary to set the triggering level such that an approximately 100% duty cycle was realized. In other words the driving signals were initiated and terminated at the average voltage as shown in Figure 7-4.

The time relationship between the hall effect device output signals and the rectified driving current to one phase may be seen in Figure 7-5. (The current is referred to as rectified since the current through the phase actually reverses every second current pulse.) The corresponding voltage pulses to the two phases are shown in Figure 7-6. The phase lag may be observed from this figure.

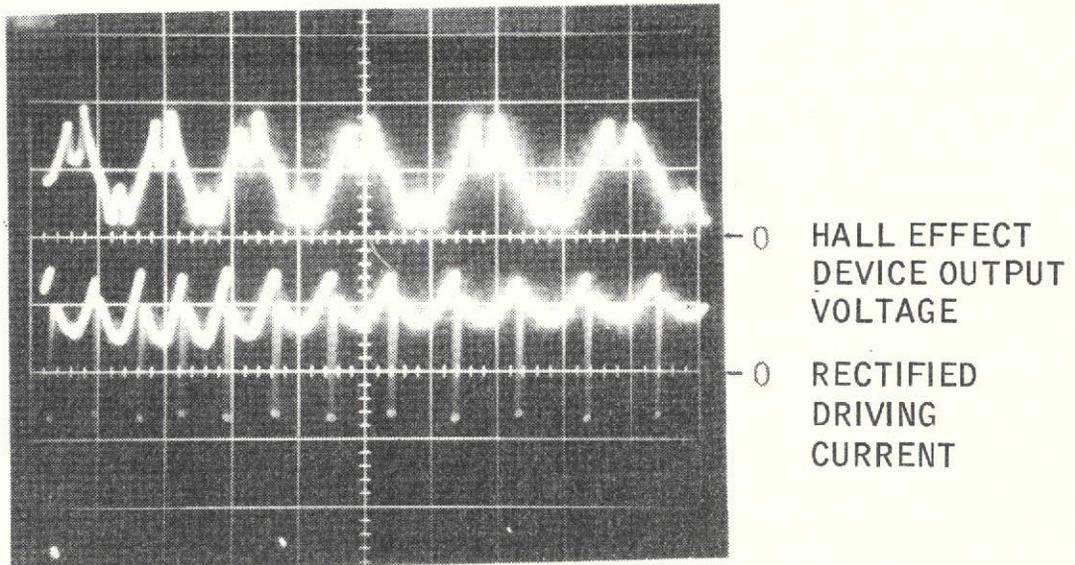
Driving of the brushless DC torque motor with a 100% duty cycle rather than with a 60% duty cycle resulted in additional power consumption of approximately 60% and in a more substantial torque ripple. It is apparent however that the magnet spacing in the rotor can be improved so as to reduce the magnitude of the dip in the hall effect device signal to thereby permit adjustment of the electronic driver to the 60% duty cycle. Thus it appears readily feasible that the originally predicted power consumption level can be met with a modified rotor.

The specific torque motor used during this development program was designed for an operating voltage of 15 volts DC. Consequently at voltages higher than 15 volts the current limiting circuit included in the electronic driver took effect and prevented the average current from exceeding a pre set value. Thus at voltages above 15 volts the stall torque is a constant and is equal to 3.5 inch pounds. The torque motor no load speed characteristics are presented in Figure 7-7. A maximum no load speed of 1320 rpm at 30 volts DC was demonstrated. The no load speed of 650 rpm at 15 volts DC compares to a vendor predicted no load speed of 720 rpm at the same voltage.



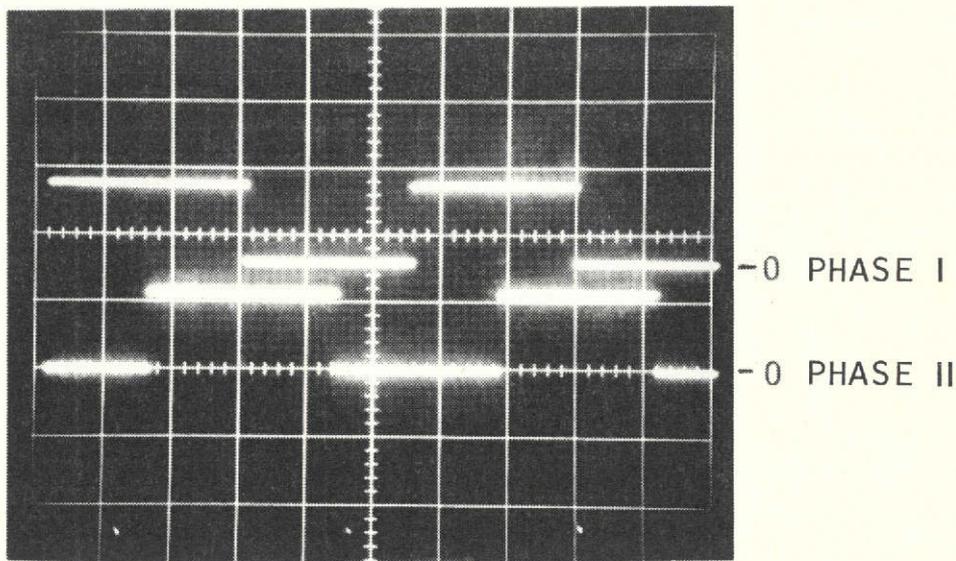
HALL EFFECT DEVICE OUTPUT VS TIME

Figure 7-4



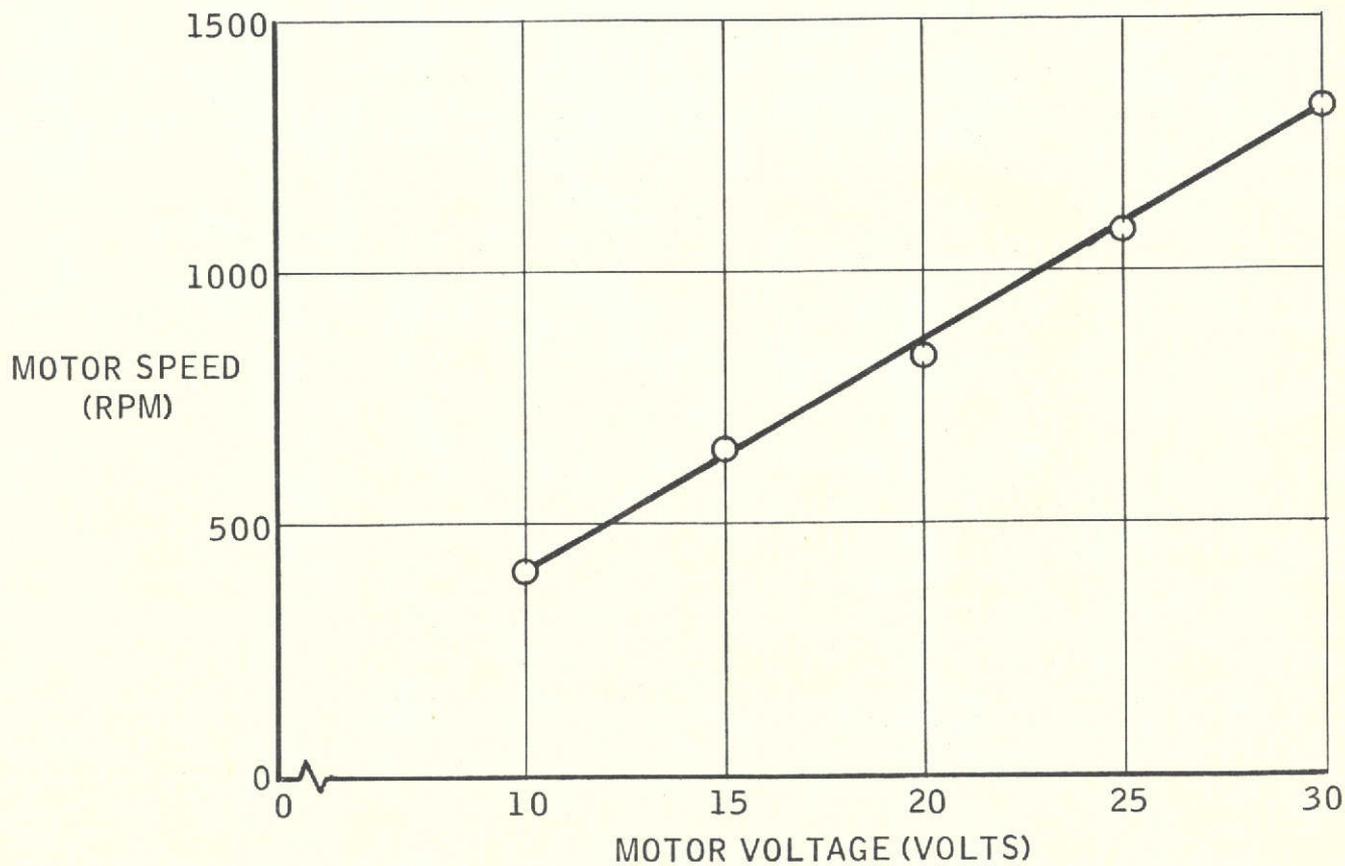
HALL EFFECT DEVICE OUTPUT VOLTAGE
AND RECTIFIED DRIVING CURRENT
VS TIME (PHASE II)

Figure 7-5



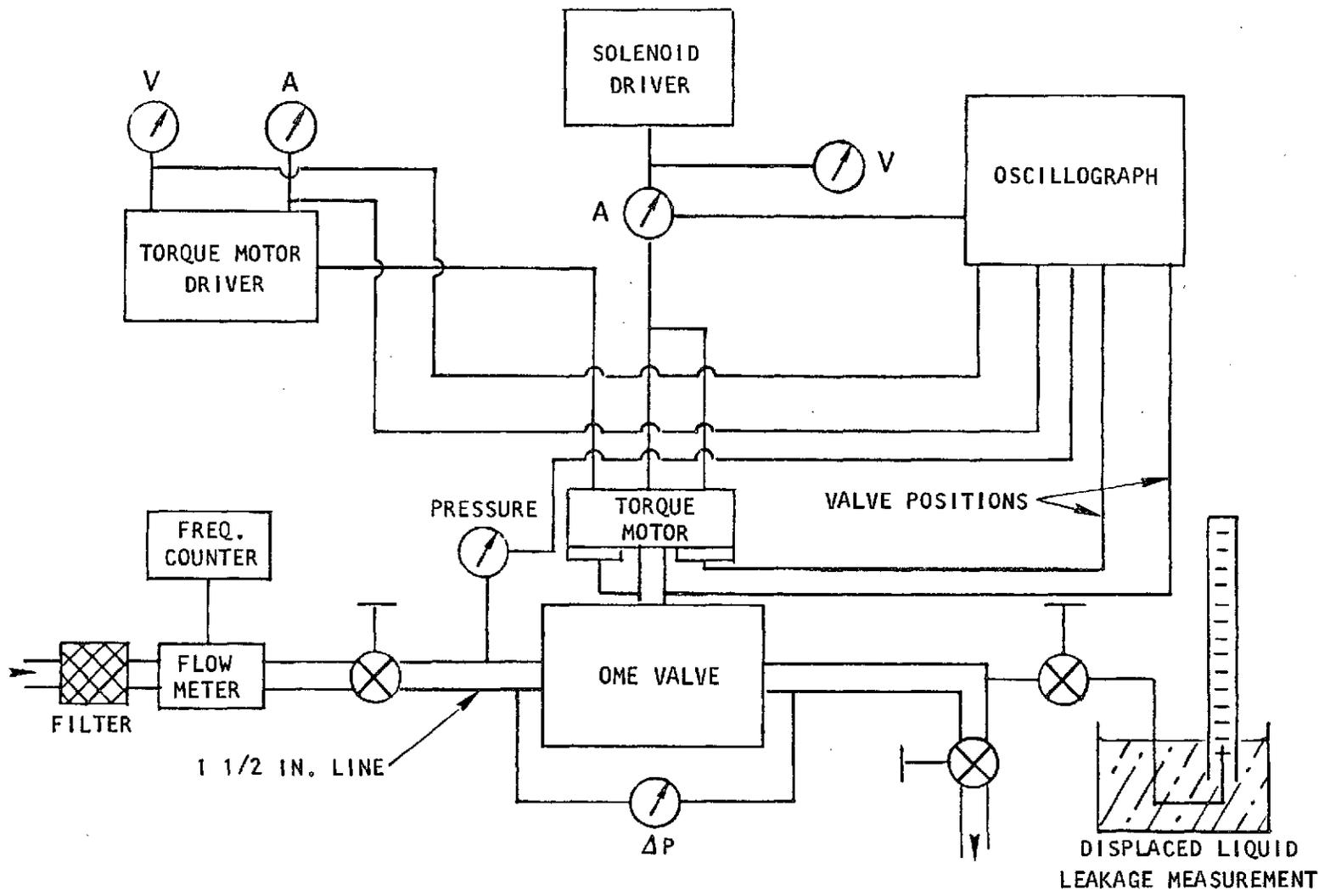
DRIVING VOLTAGE VS TIME

Figure 7-6



TORQUE MOTOR NO LOAD SPEED VS VOLTAGE

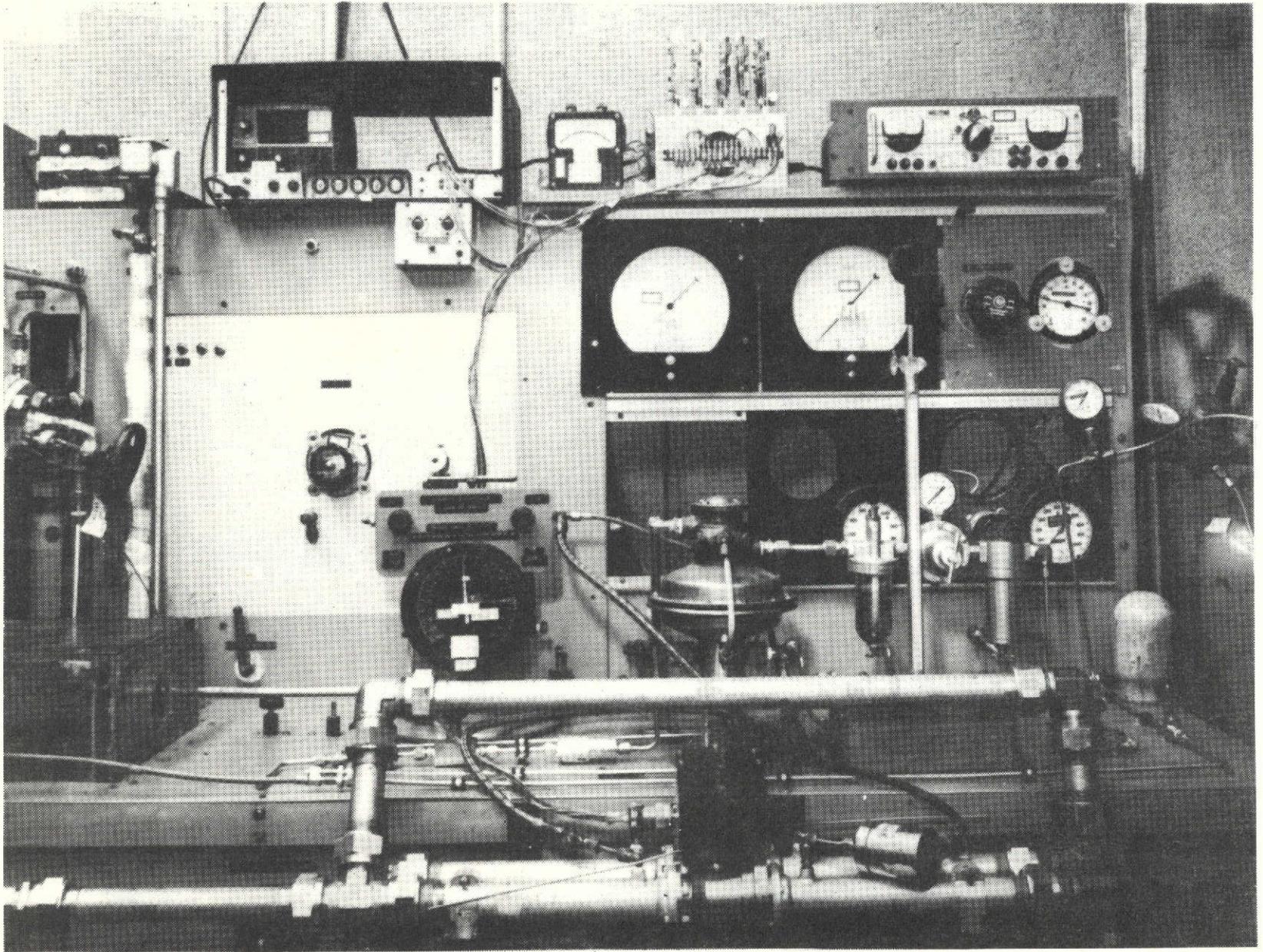
Figure 7-7



7-8

Figure 7-8

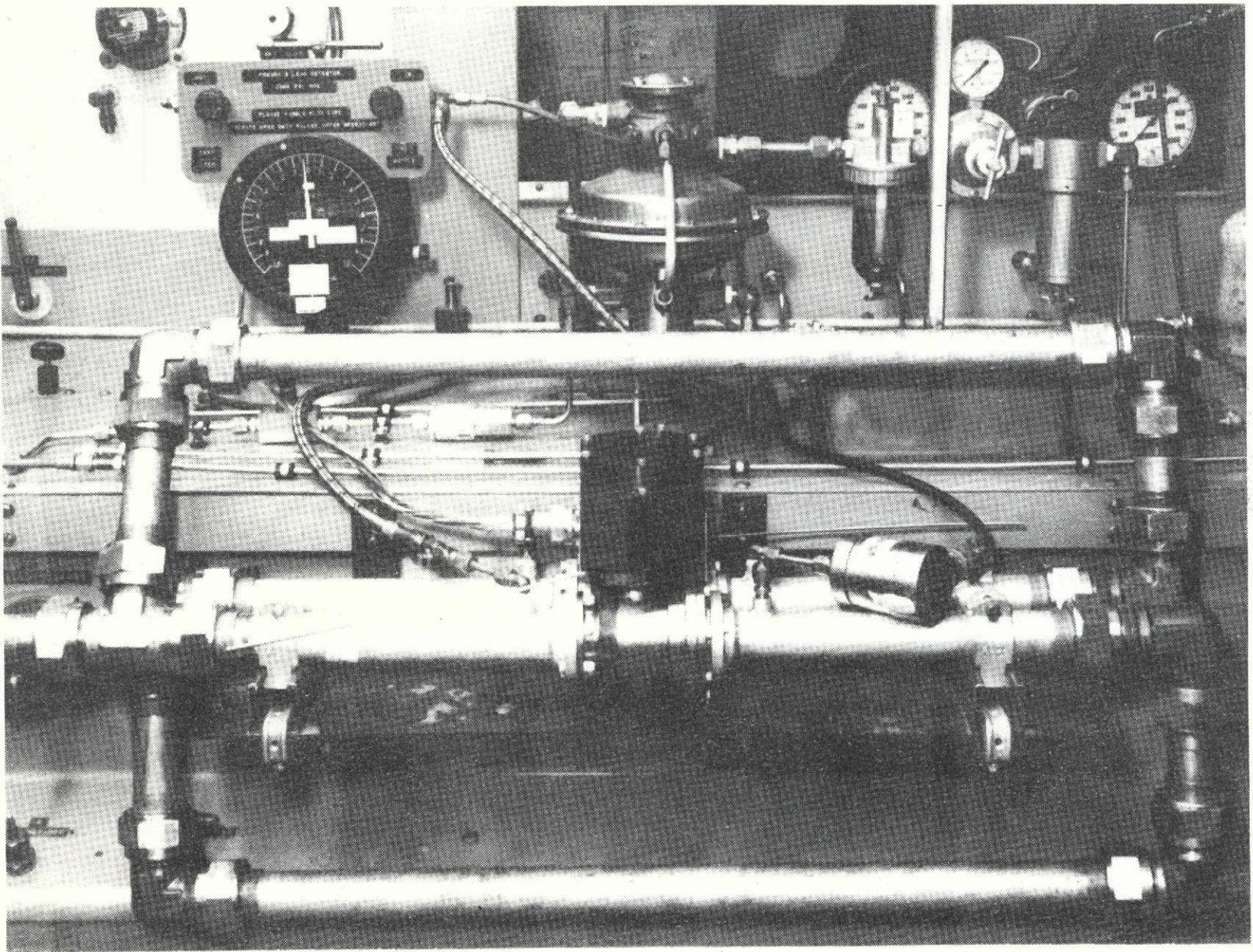
OME VALVE TEST SETUP SCHEMATIC



OME VALVE TEST SYSTEM

7-9

Figure 7-9



OME VALVE INLET AND OUTLET PLUMBING

7-10

Figure 7-10

7.2.2 Bipropellant Valve and Actuation System Tests

The bipropellant valve and actuation system was tested on the water flow bench in Building 37 at the Marquardt Van Nuys Facility. A schematic of the test set-up and a photograph are presented in Figures 7-8 and 7-9 respectively. Water for the flow loop was obtained from a pressurized tank and was dumped overboard down stream of the test item. The water flow loop included throttling valves both upstream and downstream of the test item to permit the determination of pressure drop characteristics of the OME valve and to set the water flow rates during the life cycle program. Valve upstream pressure, water tank pressure, and valve pressure drop were measured on the gauges shown in Figure 7-9. Valve upstream pressure was also recorded on oscillograph to monitor pressure surges during valve closing. Torque motor driving voltage and current as well as solenoid driving current and valve position were also recorded on the oscillograph. Valve leakage was measured by means of a positive displacement of water buret. Water flow rate was measured with a turbine type flow meter which was connected to a digital read out.

Figure 7-10 is a close-up of the plumbing immediately upstream and downstream of the test item. Since the valve was designed to feature counterflow, the oxidizer and fuel inlet and outlet plumbing had to be run accordingly and resulted in the somewhat complex setup shown. Also visible in this photograph are the 1/8" tubes emanating from the ball valve seal cavities. These tubes are used to measure fuel valve and oxidizer valve shaft seal leakage.

The pressure drop characteristics of the bipropellant ball valve were determined by capping the supply line to the fuel valve and flowing through the oxidizer valve only. The inlet pressure was set to 250 psi and the flow rate was controlled by adjusting the throttling valve downstream from the test item. The test data obtained are plotted in Figure 7-11. The design point for the ball valve was a flow rate of 5.95 pounds per second of nitrogen tetroxide with an allowable pressure drop of 0.4 psi. This corresponds to an equivalent water flow rate of 5.0 pounds per second. As evident from Figure 5-11 the pressure drop through the oxidizer valve at this flow rate is only 0.2 psi. Consequently a substantial design margin exists. The low pressure drop characteristics are believed to be due to the smooth flow path through the valve which may be observed in Figure 5-9.

An evaluation of the valve response characteristics disclosed that the opening response was excellent but that the closing response was very slow. Response characteristics as a function of operating voltage with the valve dry are plotted in Figure 7-12. Accordingly the valve opening response varies from 520 milliseconds at 35 volts DC to 1.15 seconds at 20 volts DC. The valve closing response is essentially constant regardless of voltage since it depends on the torsion spring characteristics only and is not effected by the brushless DC torque motor. As mentioned previously the valve was designed to feature a fuel lead during opening and fuel lag during closing. The opening response plotted corresponds to the fuel valve with the oxidizer valve actually being somewhat faster. Both valves reach the open position at the same time. Closing response has been plotted for the oxidizer valve and the subsequent fuel valve lag is also shown. According

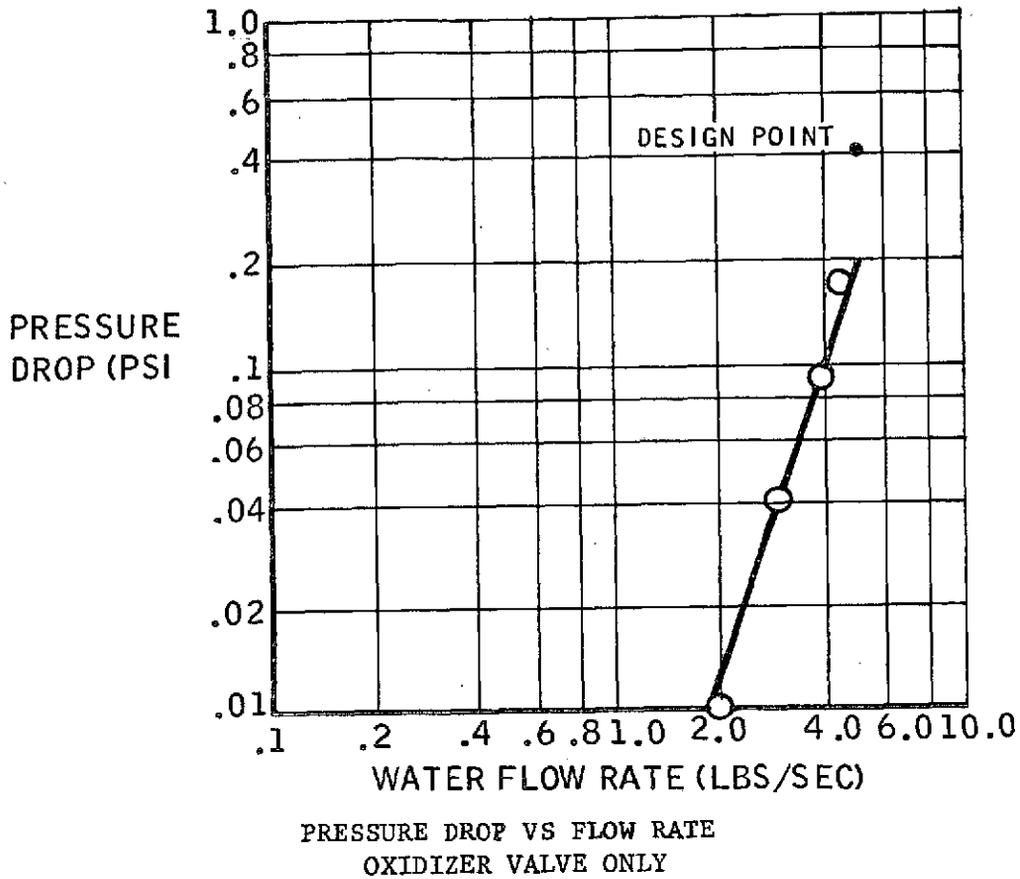


Figure 7-11

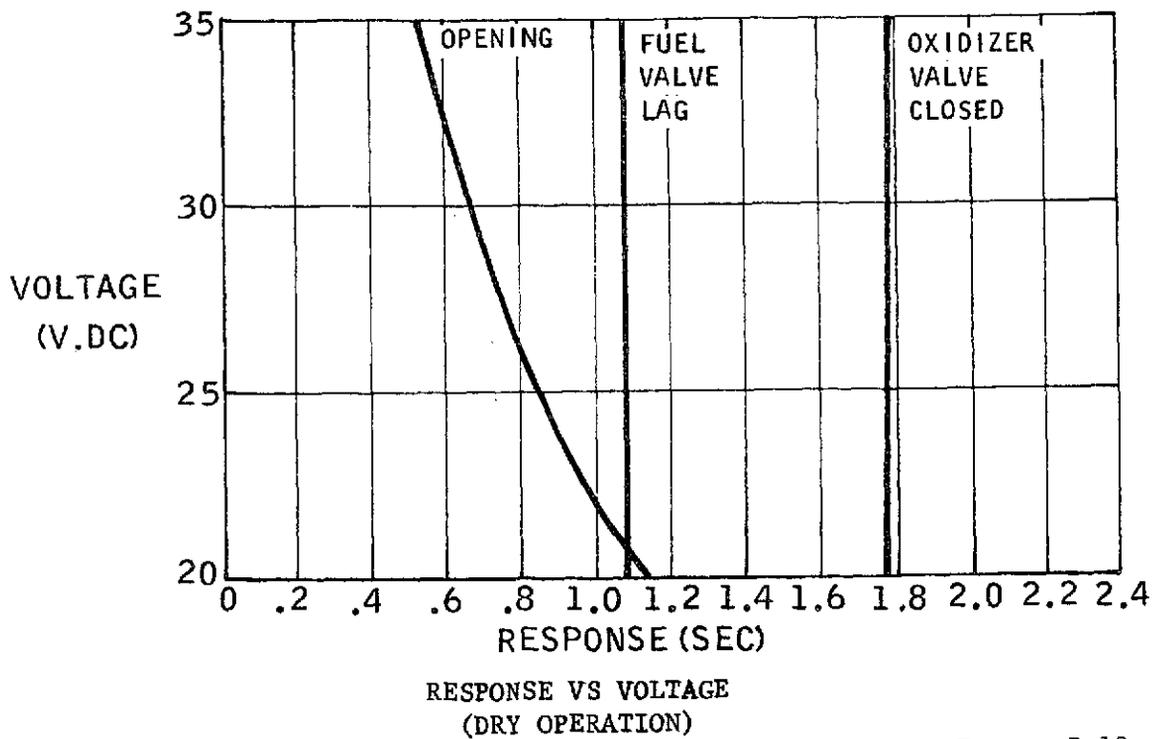
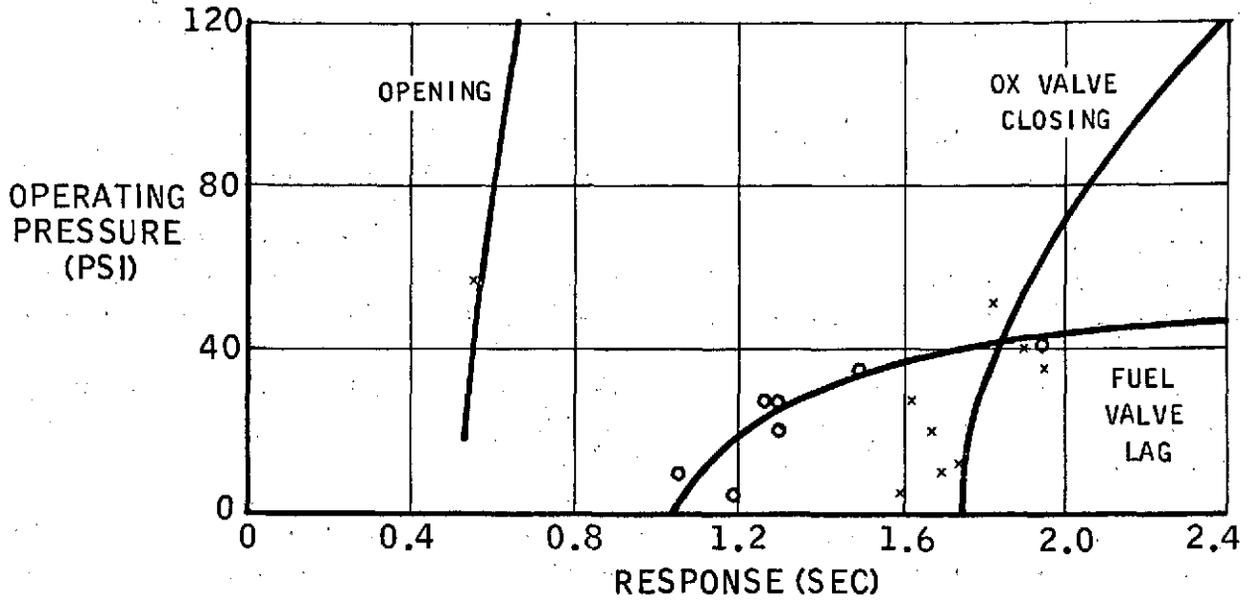
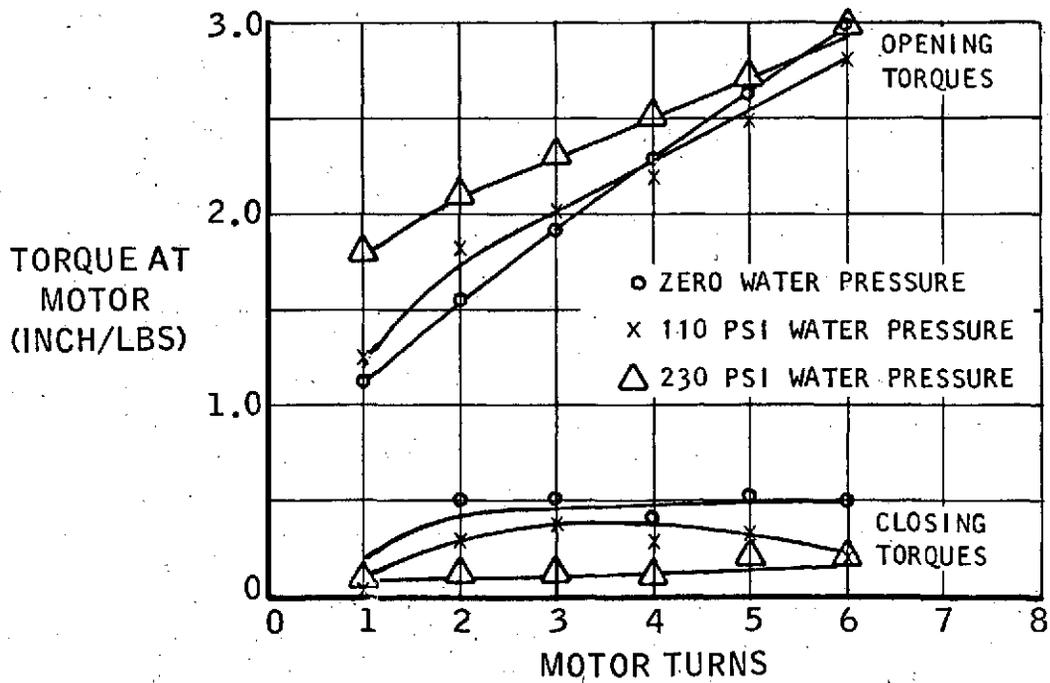


Figure 7-12



RESPONSE VS PRESSURE
WATER FLOWING, 35 VDC RUNS 6-17

Figure 7-13



TORQUE VS MOTOR TURNS
MOTOR DEENERGIZED - SOLENOIDS ENERGIZED

Figure 7-14

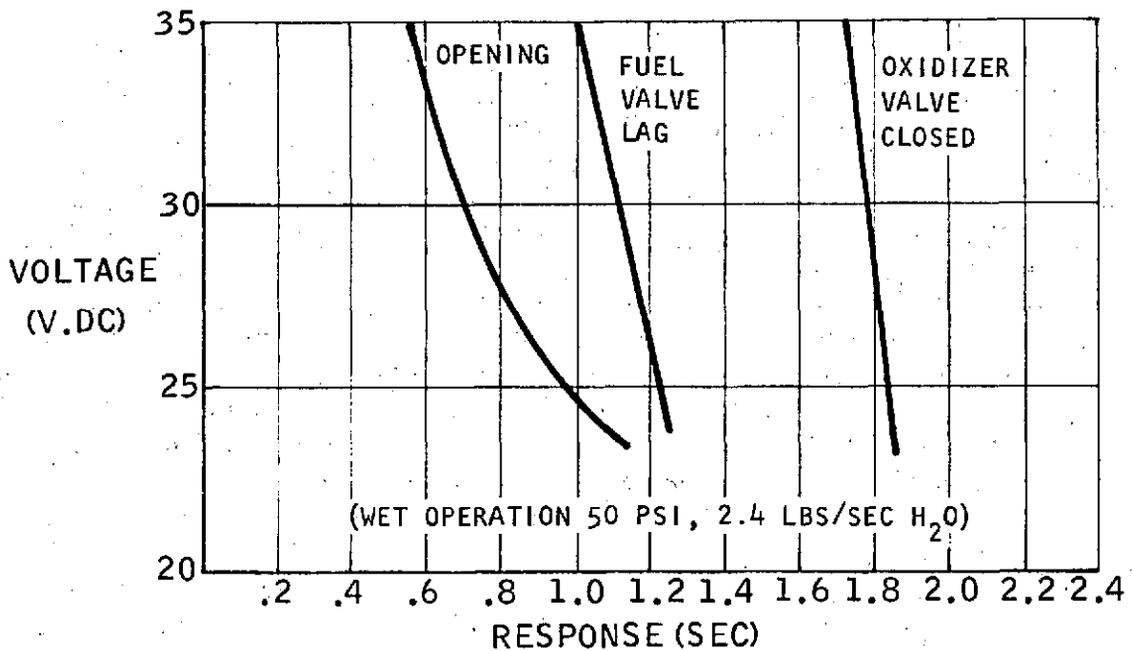
to Figure 7-12 the oxidizer valve closing response is 1.78 seconds with a subsequent fuel lag of 1.08 seconds.

Valve response as a function of operating pressure was also determined and is presented in Figure 7-13. Accordingly, the opening response decreases from 520 milliseconds at 0 pressure to 700 milliseconds at 140 psi. This response decrease is caused by the increase in friction at the shaft seals, the increased friction at the scalloped ball bearings resulting from the pressure differential across the ball acting on the ball, and the increased torque required to operate the ball resulting from the eccentric shaft design. The oxidizer valve closing response is shown to decrease from the 1.7 seconds at 0 pressure to 2.4 seconds at 120 psi. The fuel valve lag is also plotted in Figure 7-13 and indicates that the fuel valve fails to close at operating pressures in excess of 50 psi. To gain a better understanding of the slow closing response, the valve torque characteristics were reviewed.

Figure 7-14 shows the valve opening and closing torques as measured at the motor shaft at three operating pressures. These measurements were made by attaching a torque wrench to the motor shaft and reading the torque values as the valve was being opened and similarly during the closing motion, reading the torque with which the valve had to be restrained to keep it from closing. It should be noted that it takes one motor turn before the oxidizer valve is picked up to start its opening motion. Consequently, there is a step change in the torque characteristics at one motor turn resulting from the fact that below one turn, only the fuel valve torsion spring is effective while above one turn, both torsion springs are effective. Thus the data shown at one turn may be the result from driving the one or two torsion springs. The conclusion to be drawn from Figure 7-14 is that as the torque required to operate the valve during the open motion increases, the friction characteristics in the valve also increase at essentially the same rate such that the torque available for closing the valve remains constant. Thus the initial high acceleration torque required to rapidly close the valve is not available and the resultant valve closing response is relatively slow. Figure 7-14 also shows the effect of operating pressure on available closing torque. The additional seal friction at the higher operating pressure reduces the already low closing torque even further, resulting in the slower closing response shown in Figure 7-13. Indeed, below one turn the closing torque is practically negligible and the valve fails to close at higher pressures.

Marquardt's review of the increasing friction with increasing torque problem in the drive train has concluded that this is due primarily to improper design of the torsion springs. The torsion springs were designed without any clearance between adjacent coils in the installed condition. Consequently, as the springs were wound up, there is a tendency for the spring to want to grow in length and to shrink in diameter since an additional 3/4 turn is being formed (1/2 turn preload and 1/4 turn opening motion). Since the torsion springs are confined in the axial direction, this increase in spring length results in a diametral deformation and increased friction between adjacent spring coils. It is apparent that a simple redesign of the torsion springs to feature a finite spacing between adjacent coils will eliminate this problem and should result in the desired closing response characteristics.

Since the fuel valve would not close at operating pressures above 50 psi, it was decided to perform the life cycling at that pressure. It is believed that this approach is still valid in evaluating valve life cycle characteristics since the majority of cycles were being performed dry anyway and since the opening motion characteristics subject the drive train to essentially the same operating stresses that would have been incurred at higher operating pressures. The valve response characteristics during the water life cycle program are presented in Figure 7-15 as a function of voltage. During the life cycle program, all cycling was performed at 35 volts to obtain the fastest response characteristics and to thereby subject the valve drive train to the highest operating stress levels. Life cycling consisted of



RESPONSE VS VOLTAGE
(WET OPERATION)

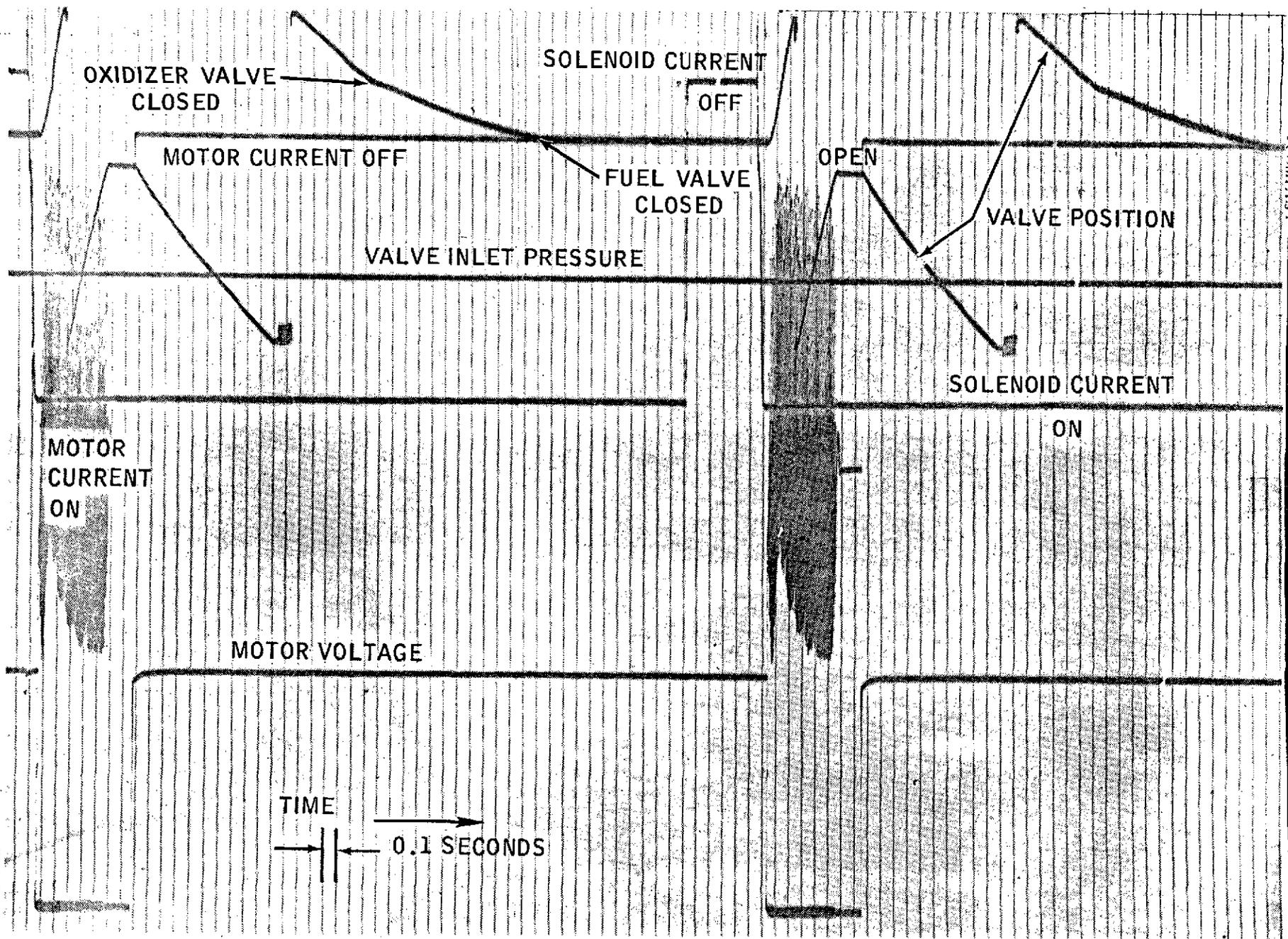
Figure 7-15

a series of "wet" life cycles, followed by a series of "dry" life cycles. All wet cycling was performed with water at 50 psi inlet pressure and all dry cycling was performed with air at 5 psi inlet pressure. A total of 10,440 cycles were accumulated. Wet cycling was performed from 1600 to 2460, 4225 to 5135, 7885 to 9920, and 10220 to 10440 cycles. All other cycles were performed dry. Cycling was performed at a rate of approximately 10 cycles per minute. Life cycle testing was interrupted periodically to permit response and leakage measurements.

A typical oscillograph trace, as taken during the life cycle program, at a speed of 1 inch per second is shown in Figure 7-16. (The normal oscillograph speed used during the life cycle program was 4 inches per second.) Two complete cycles are shown in this figure. The torque motor current trace during motor travel appears quite hashy because the current limiting circuit used in the electronic driver is active at the 35 volt operating voltage and the effect of this current limiting is such as to repeatedly turn off the current to the windings at a high frequency such that the average current to the winding is in accordance with the set current limit. The valve position trace looks a little strange because the potentiometer employed was a one turn potentiometer and was required to turn through slightly more than one turn. Consequently, during the valve travel the potentiometer would go through the normal point, causing the position signal to flip to a different level, as evident from Figure 7-16. However, the open position as well as the points where the oxidizer valve and the fuel valve reach the closed position are readily evident from the trace.

The response characteristics measured throughout the life cycle program are plotted in Figure 7-17. Opening response measurements either wet or dry or exceptionally repeatable and did not vary more than 3% throughout the life cycle program. Closing response characteristics were quite slow as mentioned previously and varied as much as $\pm 30\%$ during the program. However, considering the very small closing torque margin available, it is believed that even this much response scatter signifies very good response repeatability. As evident from Figure 7-17, there was no apparent response degradation during the life cycle program.

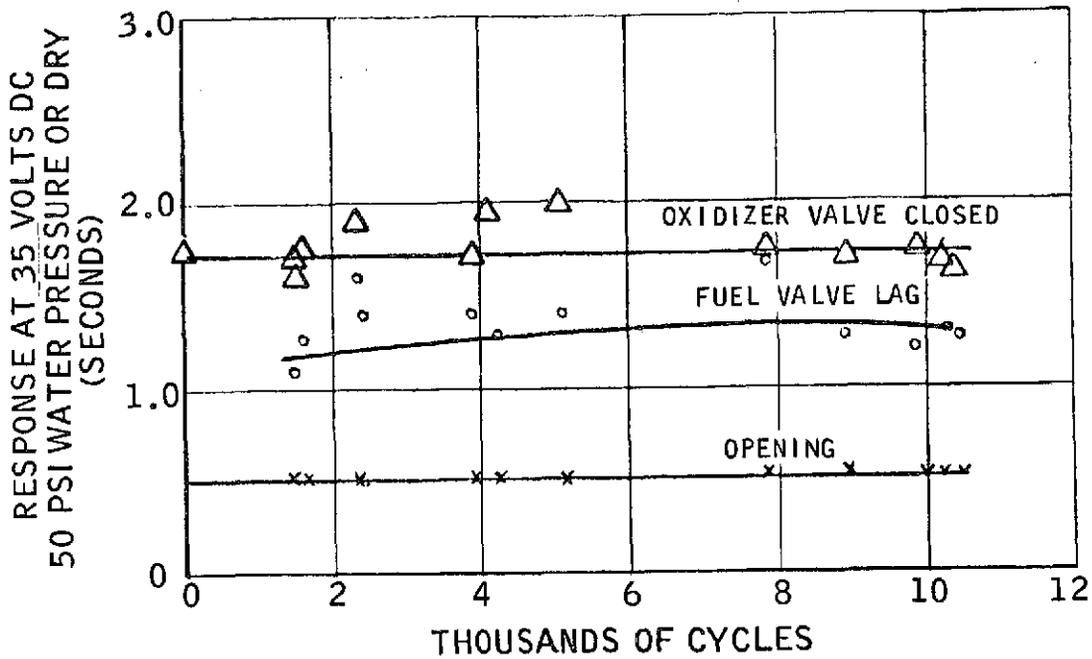
Periodic leakage measurements were made at 25 and 250 psi helium pressure at the oxidizer valve outlet, fuel valve outlet, oxidizer valve shaft seal, and fuel valve shaft seal. From the start of the life cycle program, fuel valve leakage was considered excessive and was measured at approximately 4×10^5 SCCH. The reason for this leakage is explained in a subsequent paragraph. Oxidizer valve leakage is presented in Figure 7-18 and shows some data scatter. At the initiation of the test program, as well as at approximately 9000 cycles, the valve leakage at 250 psi inlet pressure was 0 SCCH. The maximum leakage measured during the life cycle program was 1500 SCCH at 25 psi inlet pressure after approximately 5300 cycles. As evident from Figure 7-18, there was no apparent degradation of the leakage characteristics during the life cycle program. As a matter of fact, with the exception of one data point at 10000 cycles and 250 psi inlet pressure, it appears that there may be a general improvement of leakage characteristics with cycle life. This reinforces the apparent desirability of the lift-off seal design trend. Shaft seal leakage was measured to be consistently 0



OSCILLOGRAPH TRACE - LIFE CYCLE PROGRAM (2 CYCLES SHOWN)

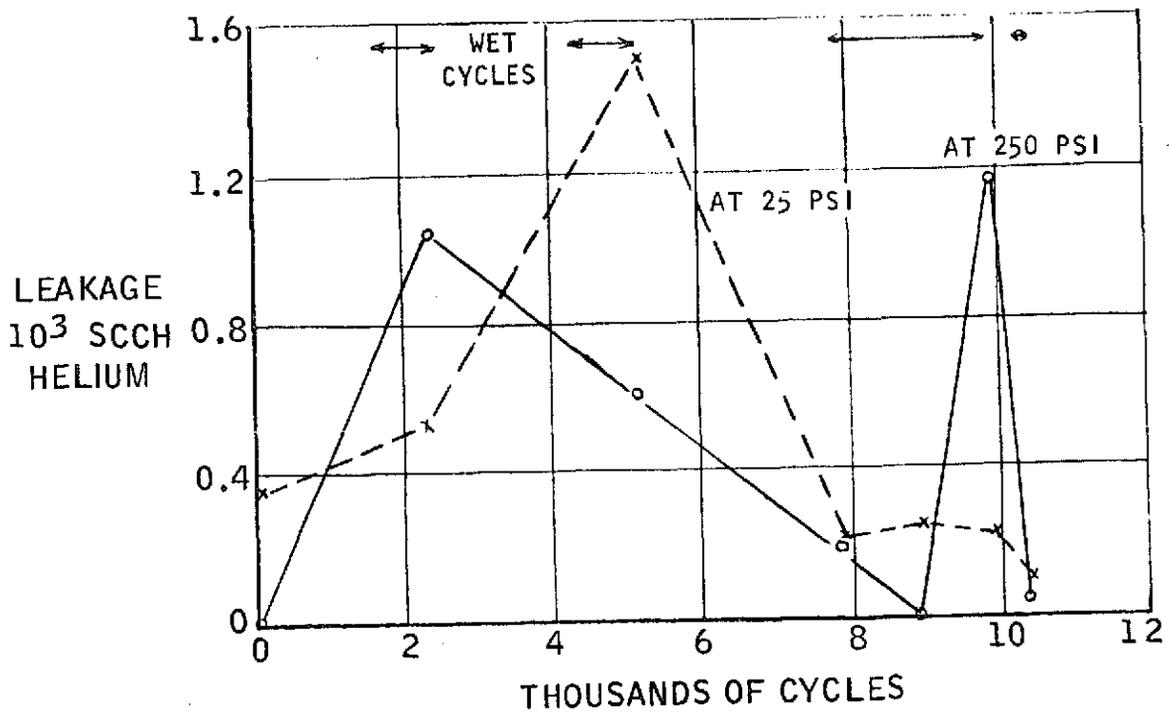
7-17

Figure 7-16



RESPONSE VS ACCUMULATED CYCLES

Figure 7-17



LEAKAGE VS ACCUMULATED CYCLES
OXIDIZER VALVE

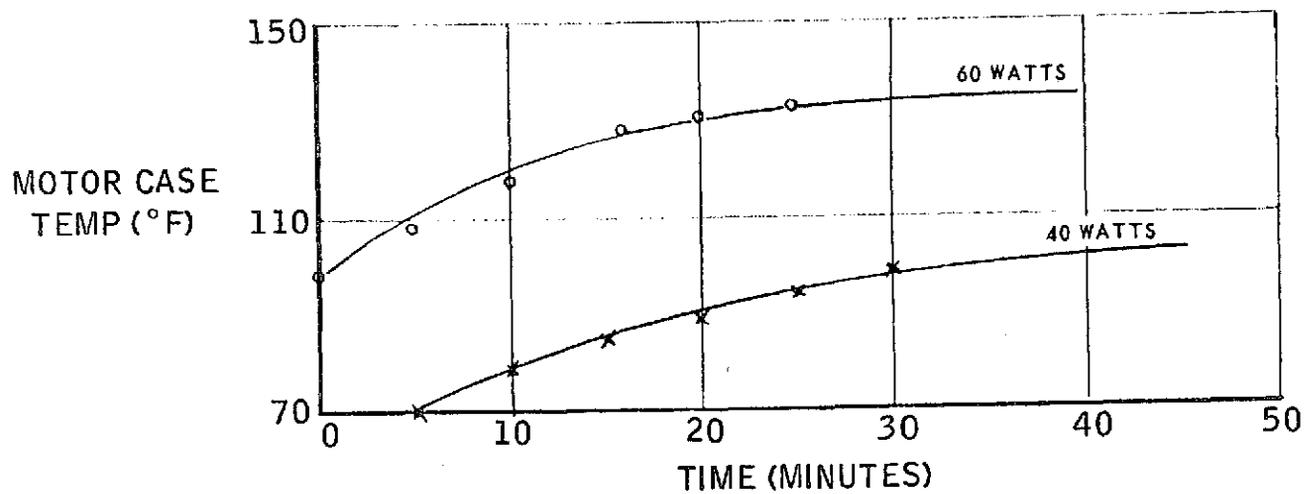
Figure 7-18

at the oxidizer valve and essentially 0 at 25 psi valve pressure at the fuel valve. Helium leakage was measured at the fuel valve shaft seals at 250 psi helium inlet pressure as follows: 388 SCCH at 2360 cycles, 20 SCCH at 7885 cycles, 130 SCCH at 8985 cycles, 220 SCCH at 9920 cycles, and 60 SCCH at 10440 cycles.

Upon completion of the life cycle program, the torque motor temperature rise during continuous on-time was determined. This data was obtained by clamping a thermocouple under one of the screws on the torque motor cover and continually energizing the torque motor and valve solenoids. The resultant data is plotted in Figure 7-19. Since the original torque motor power level goal was 40 watts, the torque motor was first energized at that power level (power to the solenoids was an additional 24 watts). During this test, the valve remained installed in the test system as shown in Figure 7-10 and the ambient temperature was 70°F. As evident from Figure 7-19, the torque motor cover temperature stabilized out at approximately 105°F. Since this temperature appeared quite low, the torque motor was also energized at a 60 watt power level and at this condition, a steady state temperature of approximately 135°F was reached. It should be noted that the prototype valve tested does feature additional material in the gear housing and torque motor cover such that a larger heat conduction path to the valve housings is provided than would be the case for the flightweight valve. Consequently, it is estimated that the flightweight valve steady state temperatures would have been somewhat higher. Nevertheless, since the maximum allowable temperature for the torque motor and hall effect devices is 210°F, a 60 watt torque motor appears very promising.

Upon completion of the test program, the bipropellant valve was disassembled step by step and each component part was inspected for possible wear or other types of damage that may have occurred during the life cycle testing. The general condition of the valve was determined to be good with minor wear (as determined by the presence of wear particles) noticeable at the Number 2 shaft seal in the oxidizer valve, at the thrust bearings of the sector gears at the gear box cover, and at the lower ball thrust bearings. More significant wear was noted at the torsion springs where they leave the aluminum retainer at the top of the valve shafts. Particularly, the fuel valve torsion spring showed severe wear between the top coil of the spring and the aluminum retainer. There was also evidence of slight wear between the adjacent turns of the torsion spring. The condition of the springs, and particularly the fuel valve springs, made it apparent that the increasing drive train friction as the torsion springs were wound up during the opening motion, which was presented previously in Figure 7-14, was primarily due to the interference between the torsion springs and the spring retainer as well as between adjacent spring coils.

Examination of the sealing closures in the two valves disclosed that the spherical sealing surfaces on the scalloped balls were rotated from the nominal closed position approximately 2.3° for the oxidizer valve and 4.3° for the fuel valve in the direction of the open position. Microscopic examination of the teflon seals disclosed the oxidizer valve seal to be in generally good condition; however, the fuel valve seal was severely gouged in one location and slightly gouged in a second location. This damage was determined to be the result of the scalloped ball over travelling in the closed direction such that the flow tube struck the sealing closure and thereby caused the described damage. The ball over-travel was attributed to a pressure surge problem that existed in the test facility when the test program was originally



TORQUE MOTOR TEMPERATURE RISE CURVE
70° F AMBIENT TEMPERATURE

Figure 7-19

initiated whereby severe water hammer effects were audible during the closure of the bi-propellant valve. Although at that time the valve inlet pressure was not recorded on an oscillograph and therefore the magnitude of the water hammer pressure was not known, it was determined that a permanent twist deformation of both fuel and oxidizer valve shafts had occurred. This deformation was observed from the fact that the valves subsequently featured gross water leakage, and upon examination of the valve outlets, that the balls had indeed over-travelled the closed position. To correct this over-travel condition, a spacer was subsequently added to the valve close position stop. The valve shafts were also stress analyzed again, and it was determined that because of the eccentric ball design, permanent twisting of the shafts could occur at pressure differentials across the ball in excess of 275 psi. The oxidizer valve leakage observed during the test program was therefore primarily attributed to the gouges in the teflon seal, and this condition was likely worsened by the fact that the ball was not fully closed, thereby resulting in lower sealing closure interface stresses and poor seat to spherical sealing surface alignment.

Post test inspection of various other valve components such as the single convolution bellows used in conjunction with the solenoid design, the gear teeth in the drive train, and various other radial and thrust bearings at the valve shafts, intermediate gears, and in the torque motor showed these components to be in excellent condition. The torque motor and the electromagnetic solenoids had performed flawlessly during the test program and appeared in excellent condition. Thus the overall appearance of the bipropellant ball valve after the life cycle program was highly encouraging and after correction of the problems observed in the preceding paragraphs, should result in a most reliable and efficient long life component.

THIS PAGE LEFT INTENTIONALLY BLANK

8.0 RECOMMENDATIONS AND CONCLUSIONS

The OMS engine shutoff valve and actuation system design and evaluation program reviewed the existing shutoff valve and actuation system concepts, particularly those used during the Apollo program, and has determined that these concepts are inadequate to satisfy the long life and zero maintenance requirements of the Space Shuttle program. Two electrical torque motor operated valve concepts, one a poppet valve and the other a ball valve, were detail designed and their performance characteristics determined. Both of these concepts offer major improvements in the areas of weight, reliability, maintainability, contamination insensitivity, and life over existing designs and are fully capable of meeting the Space Shuttle requirements. A flow fixture of the dual poppet valve sealing closure was fabricated and tested and the pressure drop requirements through this valve were substantiated. A prototype brushless DC torque motor operated bipropellant ball valve and a breadboard electronic driver were also fabricated and subjected to a performance and life cycle test program. The valve and actuation system was cycled over 10000 times and most of the performance objectives were achieved.

The test results from the ball valve and actuation system life cycle program have shown that several valve design modifications are desirable. These include a redesign of the torsion springs to eliminate contact between the adjacent coils and between the end fittings of the springs, an increase of the minimum cross section of the valve shafts to improve operating stress margins, a restudy of the tolerancing of the valve stops in the closed position to assure proper alignment of the ball with the sealing closure when the valve closes, and an increase in the torque motor output capability to improve torque margins. The breadboard electronic driver demonstrated during this program should be repackaged to a flight configuration and integrated into the valve gear housing in the cavity provided for this purpose. As part of this driver integration, consideration should be given to replacing the hall effect device sensors in the torque motor with light emitting diodes to further simplify the driving circuit. Also, the solenoid driving function which was handled by a separate pulser during this test program should be integrated with the flightweight driver. Upon completion of the ball valve modifications and the electronic driver integration, the valve should be subjected to another performance and life cycle program, including environmental testing such as vibration and propellant exposure.

The trade-off studies and feasibility testing performed during the OMS engine shutoff valve and actuation system design and evaluation program have conclusively demonstrated that electrically actuated valves of the poppet or ball type offer a 35% weight reduction compared to the pneumatic actuation system presently being pursued for this purpose on the Space Shuttle program. Furthermore, the dual poppet valve and the ball valve concepts feature major improvements with respect to reliability, contamination insensitivity, maintainability, and extended life compared to the baseline Space Shuttle approach. The major weight savings can be translated into a major cost savings on the Space Shuttle program and this possibility combined with the other attractive features of Marquardt's design concepts appear to make it highly desirable to continue with the development of at least one, if not both, of these concepts.

THIS PAGE LEFT INTENTIONALLY BLANK

9.0 REFERENCES

1. "Gaseous Oxygen - Gaseous Hydrogen Rocket Engine Injector Valve Technology" Final Report, Contract NAS 9-10086, by R. A. Lynch, December 1970.
2. "Advanced Technology for Space Shuttle Auxiliary Propellant Valves," by H. Wichmann, NASA CR-120975, May 1973.

THIS PAGE LEFT INTENTIONALLY BLANK

APPENDIX A
FORCE BALANCE
OMS ENGINE SHUT-OFF VALVE
DOUBLE POPPET TYPE

THIS PAGE LEFT INTENTIONALLY BLANK

PREPARED BY ALEX MARDERIAN

THE MARQUARDT CORPORATION

PAGE _____

CHECKED BY _____

REPORT _____

CLASSIFICATION _____

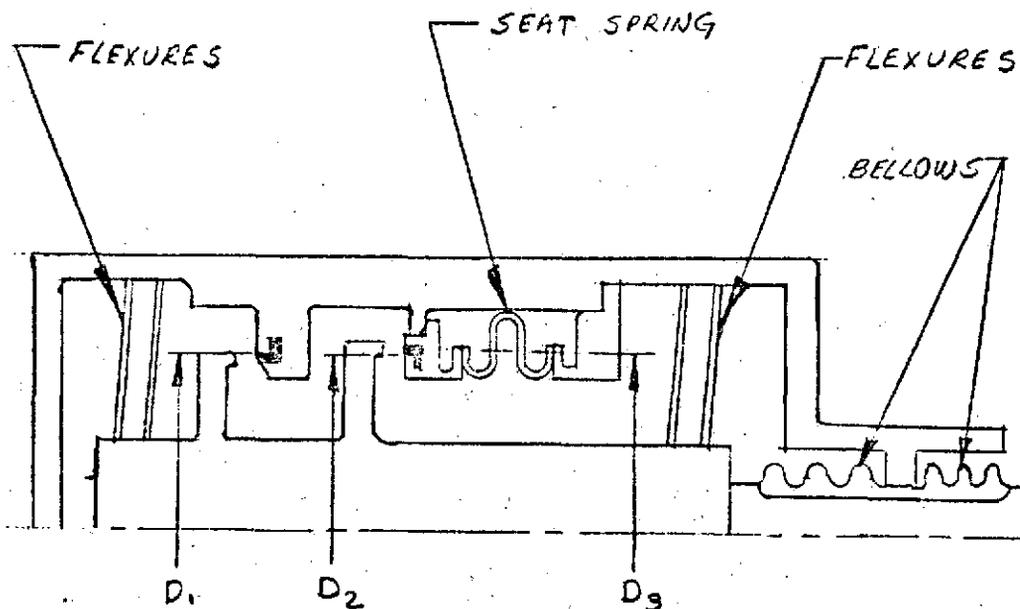
DATE JUNE 22 1973

FORCE BALANCE
OMS ENGINE SHUT-OFF VALVE
DOUBLE POPPET TYPE

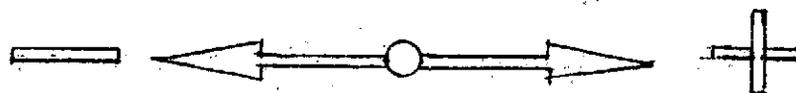
FORCE BALANCE OMS ENGINE SHUT-OFF VALVE
DOUBLE POPPET

<u>ITEM</u>	<u>PAGE NO.</u>
DIAGRAM	1
1.0 SEAT SPRING BELLOWS	2
2.0 BELLOWS, ROD SEALING	3
3.0 FLEXURES	4
4.0 FORCES ON TRANSLATING SEAT (3.45 MIN. 64.0 MAX @ 375psi)	5
5.0 SEAT DIAMETERS	6
6.0 FORCES ON STATIONARY SEAT (3.0LB MIN 71 LB MAX @ 375psi)	7
7.0 SPRING FORCE AND RATE	8
8.0 TOTAL SPRING FORCE AND RATE CLOSING THE VALVE	8
9.0 MAXIMUM FORCES REQUIRED TO OPEN VALVE	
CALCULATIONS	9,10
GRAPH	11
10.0 MINIMUM FORCES REQUIRED TO OPEN VALVE	
CALCULATIONS	12,13
GRAPH	14

FORCE BALANCE



FORCE DIRECTION



D_1 = STATIONARY SEAT DIAMETER

D_2 = TRANSLATING SEAT DIAMETER

D_3 = EFFECTIVE DIAMETER OF SEAT SPRING

POPPET TRAVEL = .13"

PRESSURES: OPERATING 250 PSI \pm 25 PSI

PROOF 375 PSI

BURST 625 PSI

FORCE BALANCE (CONT'D)

1.0 SEAT SPRING BELLOWS

METAL BELLOWS CORP
20977 KNAPP ST
CHATSORTH CA. 91311

MR. JIM WRIGLEY (SALES)
341-4900

WELDED BELLOWS ONE CONVOLUTION

MATERIAL AM350

THICKNESS = .007"

1.677 O.DIA.

SPRING RATE = 1150 LB/IN.

1.067 I.DIA.

EFFECTIVE AREA = 1.480 IN² (1.373 DIA.)

PRELOAD DEFLECTION = .001" to .005" (COMPRESSION)

OPERATING DEFLECTION (BEYOND PRELOAD) = .002" to .005"

PRELOAD = { 1.15 LB MIN.
5.75 LB MAX.

OPERATING LOAD = { 3.95 LB MIN
11.50 LB MAX.

FORCE BALANCE (CONT'D)

2.0 BELLOWS, ROD SEALING

TWO BELLOWS IN SERIES REDUNDANT
 NULL IS AT POPPET CLOSED POSITION

MINIFLEX CORP. PART NO. 515-320-30-33

RATE = 33 LB/IN (PER BELLOW)
 EFFECTIVE AREA = .125 IN²
 BURST PRESSURE = 1400 PSI.

.305 I.DIA. .003 THICKNESS
 .493 O.DIA. 14 CONVOLUTIONS
 .322 O.DIA. NECKS
 .14 NECK LENGTH
 .60 FREE LENGTH CONVOLUTIONS
 .160" MAX. ALLOWABLE DEFLECTION

TOTAL RATE = 2(33) = 66 LB/IN.

→ BELLOWS SPRING LOAD @ VALVE FULL OPEN

→ F = 66 (.130") = 9.0 LB.

<u>PRESSURE</u> psi	<u>FORCE</u> - LB
625	78.1
375	46.9
275	34.4
250	31.3
225	28.1

FORCE BALANCE (CONT'D)

3.0 FLEXURES

STARTING POINT:

USE FLEXURES DESIGNED FOR FLOW LIMITER (L14074)

RATE PER FLEXURE = 8.0 LB/IN

MAX. DEFLECTION = .250"

MEAN DIA. = 1.10"

THICKNESS = .020"

BEAM WIDTH = .200

DECREASE THICKNESS TO .015"

INCREASE MEAN DIA. TO 1.25"

INCREASE BEAM WIDTH TO 0.25"

$$\text{NEW RATE} = 8 \times \left[\left(\frac{.25}{.20} \right) \left(\frac{.015}{.020} \right)^3 \left(\frac{1.10}{1.25} \right)^2 \right] = 3.3 \text{ LB/IN}$$

(PER FLEXURE) PER FLEXURE

NEW STRESS @ .250" DEFLECTION

 $\sigma =$ EXISTING DESIGN STRESS

$$\sigma' = \sigma \left[\left(\frac{.25}{.20} \right) \left(\frac{1.25}{1.10} \right) \left(\frac{.015}{.020} \right)^3 \right]$$

$$\sigma' = 0.60 \sigma$$

USING 12 FLEXURES

$$\text{TOTAL RATE} = 40 \text{ LBS/IN}$$

(VALVE CLOSED) PRELOAD = 5.0 LBS (.13" DEFLECTION)

(VALVE OPEN) LOAD @ .125" DEFLECTION = 10 LB. (.25" TOTAL DEFLECTION)

FORCE BALANCE (CONT'D)

4.0 FORCES ON TRANSLATING SEAT

4.1 FIND MINIMUM VALUE OF Δ_2 USING MIN.

23 LBS. SEATING FORCE @ 250 PSI

$$250 \Delta_2 + .003 (1150) = 23$$

PRESSURE SPRING

$$\Delta_2 = \frac{23 - 3.45}{250} = .0783 \text{ IN}^2$$

LET: $\Delta_2 = .080''$ MIN.

$\Delta_2 = .140''$ MAX.

4.2 FORCES ON SEAT

$$F_{\text{MIN}} = P (.080) + 3.45$$

$$F_{\text{MAX}} = P (.140) + 11.50$$

(SEE PAGE 2 ITEM 1.0
FOR SEAT SPRING FORCES)

<u>P</u>	<u>F_{MIN} - LB</u>	<u>F_{MAX} - LB</u>
0	3.45	11.50
225	21.45	43.0
250	23.45	46.5
275	25.45	50.0
300	27.45	53.5
375	33.45	64.0
625	53.45	99.0

FORCE BALANCE (CONT'D)

5.0 SEAT DIAMETERS

D ₁			D ₂			D ₃	
DIAMETER INCHES	AREA IN ²	Δ AREA IN ²	DIAMETER INCHES	AREA IN ²	Δ AREA IN ²	DIAMETER INCHES	AREA IN ²
1.380	1.495	.140 MAX	1.314	1.355	.140 MAX	1.380	1.495
1.373	1.480	.110	1.321	1.370	.110	1.373	1.480
1.366	1.465	.080 MIN	1.328	1.385	.080 MIN	1.366	1.465

$$\Delta_1 = (\text{AREA } D_1) - (\text{AREA } D_2)$$

$$\Delta_2 = (\text{AREA } D_3) - (\text{AREA } D_2) = .080 \text{ IN}^2 \text{ SEE ITEM 4.1 PAGE 5}$$

P = PRESSURE

P PSI	P x Δ ₁ LBS		P x Δ ₂ LBS	
	MIN. LB.	MAX LB	MIN. LB	MAX LB
225	18.0	31.5	18.0	31.5
250	20.0	35.0	20.0	35.0
275	22.00	38.5	22.0	38.5
300	24.0	42.0	24.0	42.0
375	30.0	52.5	30.0	52.5
625	50.0	87.5	50.0	87.5

FORCE BALANCE (CONT'D)

7.0 SPRING

PRE LOAD = $\left. \begin{array}{l} 9.5 \text{ MIN} \\ 11.0 \text{ MAX} \end{array} \right\} \text{ ITEM 6.0}$

LET: RATE = 40 LB/IN.

IN OPENING VALVE THE LOAD INCREASE = ΔF
 $\Delta F = .125 (40) = 10 \text{ LB}$
5"

MAX. OPERATING LOAD = 20 - 22 LB.

8.0 TOTAL SPRING FORCES CLOSING VALVE

	VALVE OPEN	VALVE CLOSED	RATE LB/IN
SEALING BELLAWS	9.0 LB	0	66
FLEXURES	10.0	5.0	40
SPRING	21 ¹⁵	10.0	40
TOTALS	40 LB <u>34 LB</u>	15 LB	146 LB/IN

FORCE BALANCE (CONT'D)

9.0 MAX FORCES REQ'D TO OPEN VALVE

9.1 VALVE WIDE OPEN POSITION

ITEM B. TOTAL SPRING FORCES $\times 1/10 = 44$

PRESSURE FORCE ON SEALING BELLOWS $= .125 P$

$$F_0 \text{ MAX} = 44 + .125 P$$

<u>P - PSI</u>	<u>F₀ MAX - LB</u>
225	72.1
250	75.3
275	78.4
300	81.5
375	90.9

9.2 VALVE .002" FROM CLOSE POSITION

ITEM B TOTAL SPRING FORCES $= 44 - (.127 \times 146) = 25.5$

PRESSURE FORCE ON SEALING BELLOWS $= .125 P$

TRANSLATING SEAT SPRING PRELOAD $= .001(1150) = -1.0 \text{ LB.}$

ITEM 5.0 Δ_2 AREA DIFFERENTIAL (MIN.) $= -0.08 P$

ITEM 5.0 ($\Delta_1 = (1.495 - 1.385) = .11 \text{ IN}^2$) $= +.110 P$

$$F_0' \text{ MAX} = 24.5 + .155 P$$

<u>P</u>	<u>JUST CONTACTING SEAT</u>	<u>F₀' MAX (TO MOVE SEAT)</u>
225	58.1	59.1
250	62.3	63.3
275	66.1	67.1
300	70.0	71.0
375	81.6	82.6

A-11

FORCE BALANCE (CONT'D)

9.3 VALVE FULLY CLOSED.

$$\text{ITEM 8, TOTAL SPRING FORCES} = 44 - .13(146) = 25.0$$

$$\text{PRESSURE FORCE ON SEALING BELLOWS} = .125 P$$

$$\text{TRANSLATING SEAT SPRING} = .003(1150) = -3.45 \text{ LB}$$

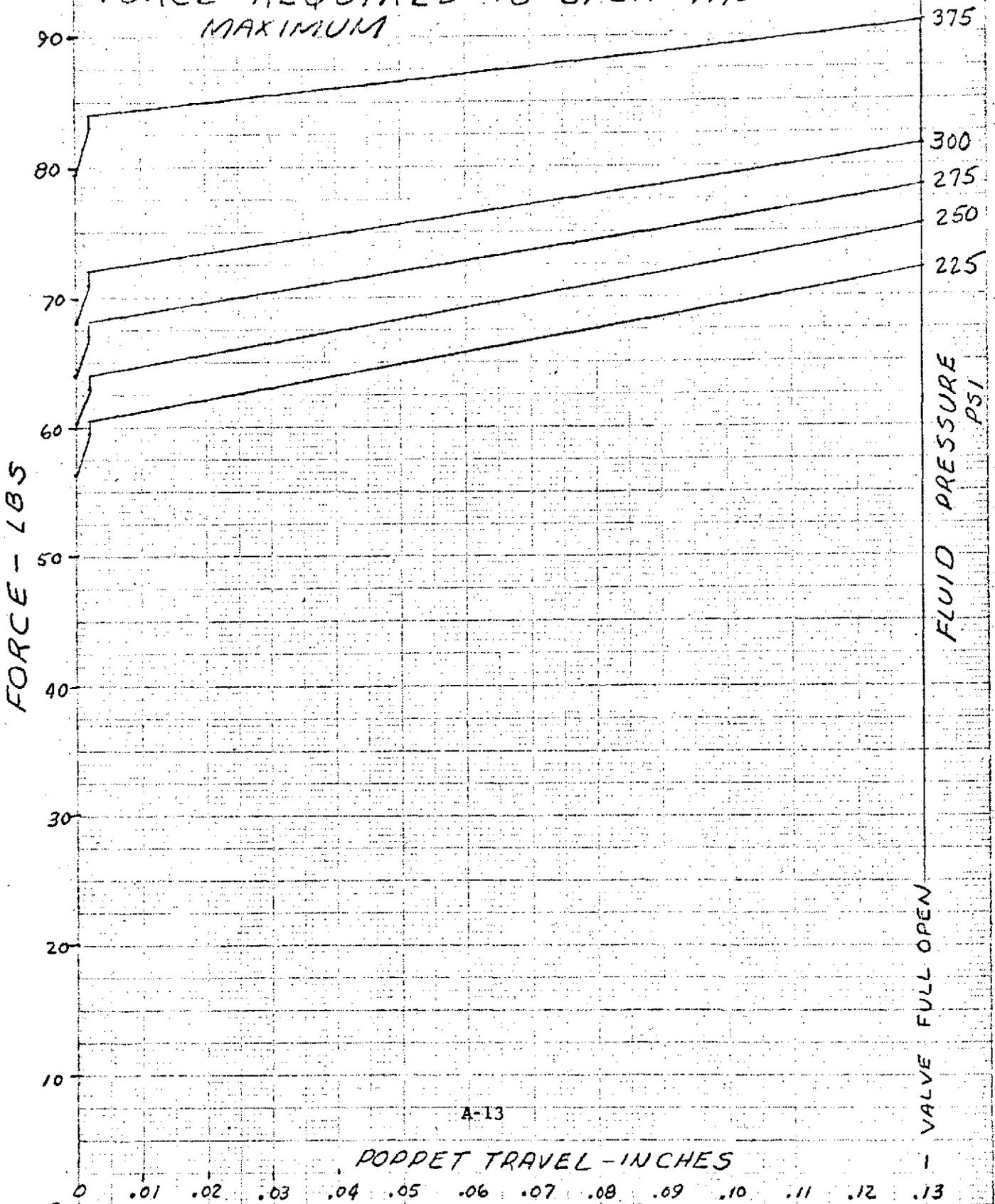
$$\text{ITEM 5.0 } \Delta_2 \text{ (MIN.) DIFFERENTIAL AREA} = -.08 P$$

$$\text{ITEM 5.0 } \Delta_1 (= .110 \text{ IN}^2 \text{ SEE ITEM 9.2}) = +.110 P$$

$$F_0'' \text{ MAX} = 21.5 + .155 P$$

<u>P</u> PSI	<u>F₀'' MAX</u>
225	56.4
250	60.3
275	64.1
300	68.0
375	79.6

FORCE REQUIRED TO OPEN VALVE MAXIMUM



A-13

POPPET TRAVEL - INCHES

FORCE BALANCE (CONT'D)

10.0 MINIMUM FORCES REQ'D TO OPEN VALVE

10.1 VALVE IN WIDE OPEN POSITION

ITEM B TOTAL SPRING FORCES = 40

PRESSURE FORCE ON SEALING BELLOWS = .125 P

$$F_0 \text{ MIN} = 40 + .125 P$$

<u>P</u> <u>PSI</u>	<u>F₀ MIN</u> <u>- LB</u>
225	68.1
250	71.3
275	74.7
300	77.5
375	86.9

10.2 VALVE .005" FROM CLOSED POSITION

ITEM B TOTAL SPRING FORCE = 40 - (.125 x 146) = 22

PRESSURE FORCE ON SEALING BELLOWS = .125 P

TRANSLATING SEAT PRELOAD = - 6.0 LB

ITEM 5.0 Δ₂ MAX (DIFF. AREA) = -.14 P

ITEM 5.0 Δ₁ (= 1.465 - 1.355 = .110) = +.11 P

CASE A. JUST CONTACTING SEAT $F_0' \text{ (MIN)} = 22 + .095 P$

CASE B. START TO MOVE SEAT $F_0' \text{ MIN} = 16 + .095 P$

<u>P</u> <u>PSI</u>	<u>F₀' MIN</u> <u>CASE B - LB</u>	<u>F₀' MIN</u> <u>CASE A - LB</u>
225	37.4	43.4
250	39.8	45.8
275	42.1	48.1
300	44.5	50.5
375	51.6	57.6

FORCE BALANCE (CONT'D)

10.3 VALVE FULLY CLOSED

ITEM 8 TOTAL SPRING FORCE = $40 - (.13 \times 146) = 21$

PRESSURE FORCE ON BELLOWS = $.125 P$

TRANSLATING SEAT SPG $(.000 \times 1150) = -11.5 \text{ LB}$

ITEM 5.0 $\Delta_2 \text{ MAX} = -.14 P$

ITEM 5.0 $\Delta_1 (= .110 \text{ SEE ITEM 10.2}) = +.11 P$

$F_0'' \text{ MIN}_1 = 9.5 + .095 P$

P - PSI

$F_0'' \text{ MIN}_1$
LB

225

30.9

250

33.3

275

35.6

300

38.0

375

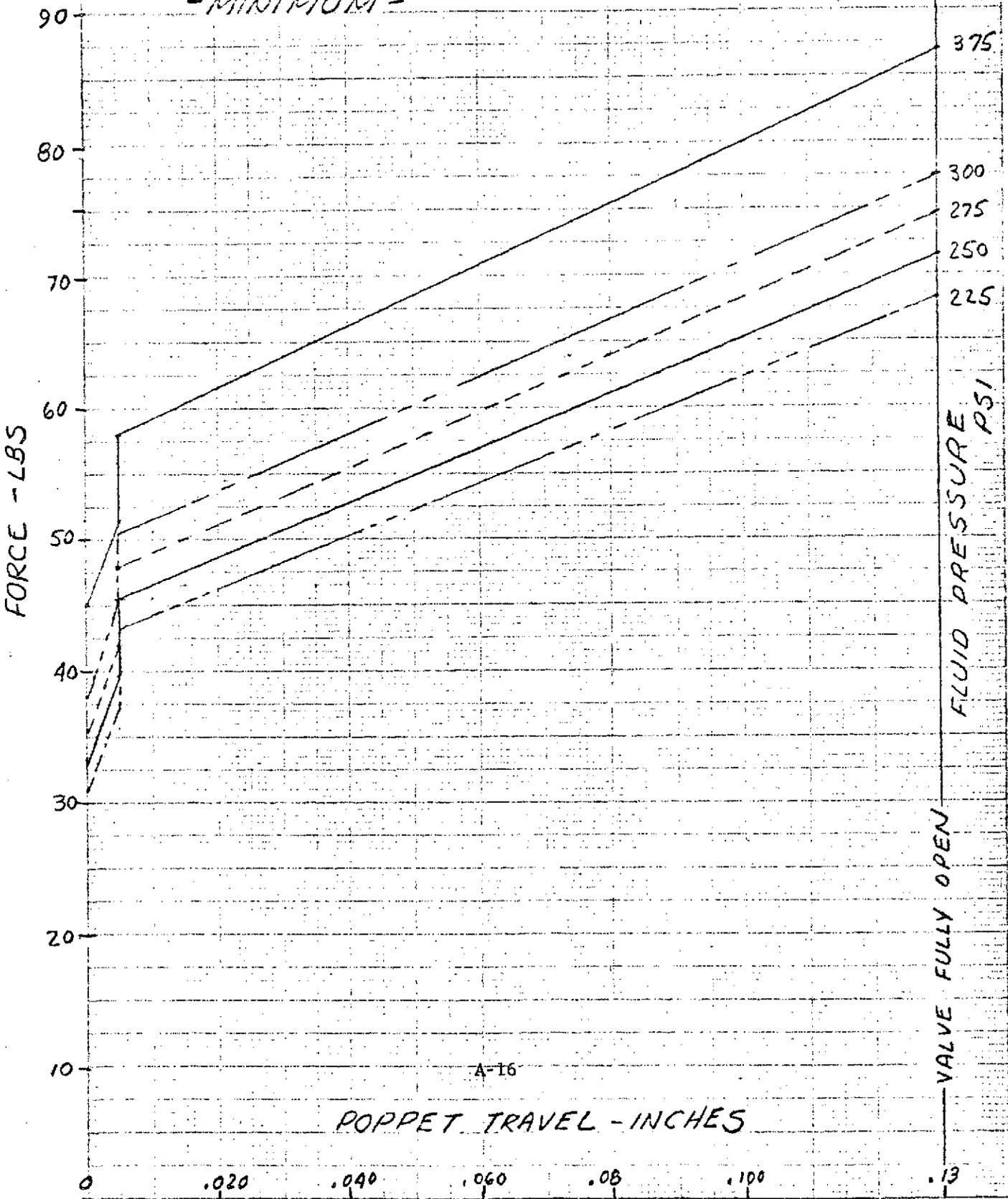
45.1



PREPARED BY ALEX MAROERIAN
CHECKED BY

PAGE 14 OF 14
DATE JUNE 22 1973

FORCE REQUIRED TO OPEN VALVE - MINIMUM -



A-16

POPPET TRAVEL - INCHES

NOT TO SCALE UNLESS LA 472
1" = 1000 INCHES

THE

Marquardt
COMPANY

VAN NUYS, CALIFORNIA

APPENDIX B

BALL VALVE STRESS ANALYSIS

THIS PAGE LEFT INTENTIONALLY BLANK

DICKENS

BALL VALVE

N-65880-20

22 OCT 1973

2 OF 4

OME BALL VALVEBELLOWS - SOLENOID LIFT OFFPRESSURE STRESS

Inco 718

t = .006FROM BELLOWS EQUATION

$$\sigma = \frac{P(o-s-t)^2}{20(t^3)} = \frac{250(.494)^2}{20(.006)^2}$$

$$\begin{aligned} o &= 1.9375 \\ s &= 1.437 \\ t &= .006 \end{aligned}$$

$$= \underline{84,800 \text{ psi}} \quad \text{for } 250 \text{ psi}$$

BELLOWS DEVELOPED LOAD

$$\text{MEAN DIAM} = (1.9375 + 1.437) / 2 = 1.6875$$

$$\text{LOAD} = p A = 250 \frac{\pi}{4} (1.6875)^2 = \underline{560 \#}$$

$$\text{SEAL DIAM} = 1.83 \text{ IN}$$

NET LOAD HOLDING BELLOWS

$$P = 250 \frac{\pi}{4} [(1.83)^2 - (1.6875)^2] = \underline{10 \#}$$

STRESS DUE TO PRELOAD = 15#

$$\begin{aligned} \sigma &= \frac{P (7.5 \times 10^6) (o-s-t)}{E (o+t) t^2} = \frac{15 (7.5 \times 10^6) (.494)}{30 \times 10^6 (3.375) (.006)^2} \\ &= \underline{15200 \text{ psi}} \end{aligned}$$

$$\text{RATE} = \frac{4.3 E (o+t) t^3}{(o-s-t)^3 N} = \frac{4.3 \times 30 \times 10^6 (3.375) (.006)^3}{(.494)^3 \times \frac{1}{2}} = \underline{1570\%}$$

$$\Delta = 15 / 1570 = \underline{.0096 \text{ IN}}$$

DICKENS

BALL VALVE

N-55880-20

22 Oct 1973

3 OF 4

OME BALL VALVEBELLOWS - SOLENOID LIFT OFF

INCO 718

AT VALVE CLOSED POSITION

$$\text{PRELOAD} = 15 \# , \sigma = 15200 \text{ psi}$$

STRESS DUE TO PROOF PRESSURE = 375 psi

$$\sigma = \frac{375}{250} \times 84800 \text{ psi} = 127200 \text{ psi}$$

$$\text{TOTAL } \sigma = 127200 + 15200 = 142400 \text{ psi}$$

$$\text{INCO 718 - H.T.A. } F_{ty} = 150,000 \text{ psi MIN. (MIL-HANDBOOK 5A)}$$

$$\text{AT WELD } F_{ty} = .9 \times 150,000 = 135000 \text{ psi}$$

$$F_{by} = 1.20 \times 135000 = 162000 \text{ psi (PLASTIC BENDING)}$$

$$\text{Proof M.S.} = \frac{162000}{142400} - 1 = .14$$

STRESS DUE TO MAX. OPERATING PRESSURE

$$P = 250 \text{ psi} , \sigma_p = 84800 \text{ psi}$$

$$\text{TOTAL } \sigma = 84800 + 15200 = 100,000 \text{ psi CLOSED}$$

STRESS WHEN OPEN

$$P = 0 \text{ psi} , \text{LOAD} = \frac{(.00967 - .006)}{.0096} \times 15 = 24.4 \#$$

$$\sigma_{\text{LOAD}} = \frac{24.4}{15} \times 15200 \text{ psi} = 24700 \text{ psi OPEN}$$

$$\text{MEAN STRESS} = \frac{100000 + 24700}{2} = 62350 \text{ psi}$$

$$\text{ALT STRESS} = \pm 37650 \text{ psi}$$

OME BALL VALVEBELLOWS-SOLENOID LIFT OFFOPERATING CONDITION LIFEFOR WELD JOINT,

DIVIDE CALCULATED STRESSES BY .90,
TO OBTAIN EQUIVALENT WROUGHT MAT'L
LIFE STRESSES.

$$\text{MEAN STRESS} = 62350 / .9 = 69250 \text{ psi}$$

$$\text{ALT STRESS} = \pm 37650 / .9 = \pm 41900 \text{ psi}$$

FROM MIL-HAND-BK 5A
FIG 6.3.8-2-8(a)

$$N \approx 10^5 \text{ cycles}$$

$$\text{M.S.} = \frac{10^5}{10^4} - 1 = \text{LARGE}$$

BURST PRESSURE CALCULATION

$$\text{PRELOAD } \sigma = 15200 \text{ psi}$$

$$F_b = 1.5 \times 160000 \times .9 = 243000 \text{ psi}$$

$$\text{MAX PRESSURE } \sigma = 243000 - 15200 = 227800 \text{ psi}$$

$$\text{MAX PRESSURE} = \frac{227800 \times 250}{84800} = 672 \text{ psi}$$

$$\text{M.S.} = \frac{672}{625} - 1 = .07$$

BALL VALVE - OMBTORSION SPRING

MEAN D = 1.375

WIRE d = .092 - CR65 302

TORQUE = 12 IN-LBS PRELOAD

= 18.5 IN-LBS AT 90° STROKE

RATE = $\frac{(18.5 - 12)}{90} 360 = 26 \text{ IN-LBS/TURN}$

NUMBER OF TURNS

$$M_a = \frac{E d^4}{10.8 D_m N}$$

$$N = \frac{E d^4}{10.8 D_m M_a} = \frac{28 \times 10^6 (.092)^4}{10.8 (1.375) 26} = \underline{5.15 \text{ ACTIVE COIL}}$$

MAX STRESS IN COIL

$$M_{\text{MAX}} = 18.5 \text{ IN-LBS}$$

$$S = \frac{32 M}{\pi d^3} = \frac{32 (18.5)}{\pi (.092)^3} = 242,000 \text{ PSI}$$

$$F_b = F_{T0 \text{ WIRE}} = 240,000 \text{ PSI FOR}$$

$$\underline{\underline{M.S. = \frac{240}{242} - 1 = .00}}$$

ACTUAL STRESS IN WIRE = 242,000 / PLASTIC BENDING MOD

PLASTIC BENDING MOD. = $\sqrt{1.69} = 1.3$

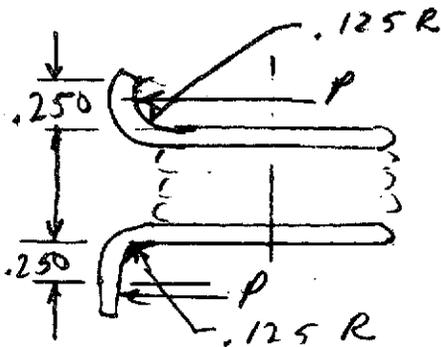
$f_b = 242 \text{ K} / 1.3 = 186 \text{ K}$

DICKENS

BALL VALVE

23 OCT 1973

2

BALL VALVETORSION SPRINGBENDING ON ENDS

$$P = \frac{T_{\text{MAX}}}{M_D/2}$$

$$P = \frac{18.5 \times 2}{1.375} = 26.9 \# \text{ LIMIT}$$

$$M_{\text{MAX}} / \text{END} \approx 26.9 \times (.30 + .092/2) \\ \approx 9.4 \text{ IN} \#$$

$$\text{RATIO OF CURVATURES} = \frac{R}{C} = \left(\frac{.125 + .092/2}{.092/2} \right) = 3.72$$

$$\text{STRESS CONCENTRATION } K = 1.25$$

$$S = 32 M K / \pi d^3 = \frac{9.4 \times 1.25 \times 242000}{18.5}$$

$$= 154000 \text{ psi}$$

$$\text{M.S.} = \frac{240}{154} - 1 = .56$$

BALL VALUE
TORSION SPRING

302 LBES

FATIGUE - 10⁴ cycles

MAX STRESS IN COIL = 186000 psi ACTUAL

MIN STRESS = $\frac{12}{18.5} \times 186000 = 121000$ psi

MEAN STRESS = $(186 + 121)K / 2 = 150350$ psi

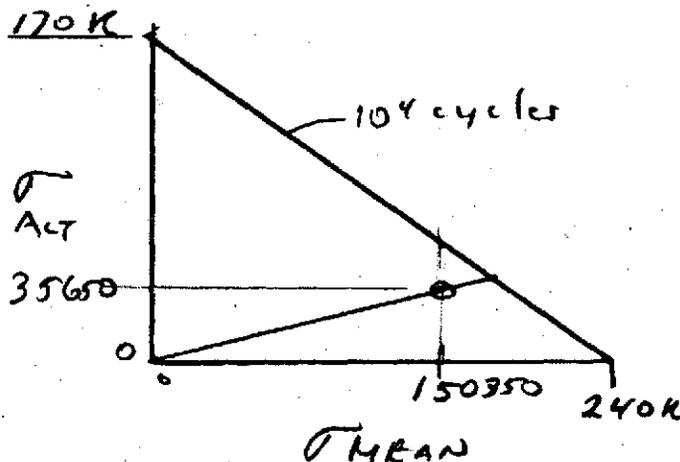
ALT. STRESS = $186000 - 150350 = \pm 35650$ psi

FROM AVAILABLE FATIGUE DATA, FEND 10⁴ cycles

= .70 FTS = $.70 \times 240,000 = 170,000$ psi

(REF JOURNAL pg 210)

GOODMAN DIAGRAM



SCALE 1" = 100KSI

10⁴ cycle FATIGUE

M.S. = $\frac{1.83}{1.56} - 1 = .17$

PREPARED BY

THE MARQUARDT COMPANY

REPORT ONE

DICKENS

BALL VALVE

CHECKED BY

CLASSIFICATION

DATE

PAGE

N-65880-20

3/06/7/923

1 OF 4

BALL VALVEBELLEVILLE SPRINGRELATIVE THERMAL EXPANSION446 COIL HOUSING AND ALUM. OUTER HOUSINGLET $\Delta T = 100^\circ F$ FOR 100° RISE IN 446 - $\alpha = 5.6 \times 10^{-6}$

$$\Delta L = 5.6(100)(.75)10^{-6} = .000420''$$

FOR 100° RISE IN AL. ALLOY $\alpha = 12.9 \times 10^{-6}$

$$\Delta L = 12.9(100)(.75)10^{-6} = .00097''$$

$$\text{LET MAX STROKE ON SPRING} = .010 + .001$$

$$= \underline{\underline{.011''}}$$

$$\text{PRELOAD} = 375 \frac{\pi}{4} (3.125)^2 = \underline{\underline{2860 \# \text{ AT PROOF}}}$$

$$\text{OD SPRING} = 3.25$$

$$\text{ID " } = 2.75$$

$$\frac{R}{P} = \frac{3.25}{2.75} = 1.18$$

FOR INCO 718 LET $F_{TY} = 150,000 \text{ PSI} = S_n$

$$S_n = 150,000 = \frac{.96 P}{t^2}$$

$$t^2 = \frac{.96(2860)}{150,000} = 0.183$$

$$t = \underline{\underline{.135 \text{ MIN}}} \rightarrow .140$$

DICKENS

BALL VALVE

N-65880-20

31 Oct 1973

3 OF 4

BALL VALVE

BELLEVILLE SPRING

FOR OD = 3.25, R = 1.625

$R_p = 1.18$

$t = .150, t^2 = .0225, t^3 = .003375, t^4 = .00050625$

STROKE = .011 AFTER PRELOAD

$$C_1 = PR^2 / CE^4, P = 2860 \text{ \# preload}$$

$$C_1 = .182 \times \left(\frac{.14}{.15} \right)^4 = .137$$

$$\text{FOR } h/c = .3, \frac{\delta}{c} = .13, \delta = .13 \times .15 = .0195$$

$$\text{TOTAL } \delta = .0195 + .011 = .0305 \text{ MAX}$$

$$\frac{\delta}{c} = \frac{.0305}{.15} = .203$$

$$C_1 = .205 \text{ FIG 9}$$

$$P_{MAX} = \frac{.205}{.137} \times 2860 = 4280 \text{ \#}$$

STRESSES

$$S_{T1} = \frac{1.1 E \delta C}{R^2} \left[-C_2 \left(h - \frac{\delta}{2} \right) + C_3 t \right]$$

C = 3.6

C₂ = 0

C₃ = 1.05

$$= \frac{1.1 \times 30 \times .0305 \times 3.6}{2.64} \left[1.05 \times .15 \right]$$

$$= 1.37 \times 10^6 (.1575)$$

$$= 216000 \text{ PSI}$$

$$F_{BY} \text{ INCO 718, HTA} = 1.5 \times 150,000 = 225000 \text{ PSI}$$

$$\text{M.S.} = \frac{225}{216} - 1 = .04$$

BALL VALVEBELLEVILLE SPRINGAT MAX LOAD = 4280 $R/H = 1.18$

$$f = .0305 \quad h = .3 \times .15 = .045''$$

$$S_{T2} = \frac{1.1 E S C}{R^2} \left[C_4 \left(h - \frac{f}{2} \right) + C_5 t \right]$$

$$= 1.37 \times 10^6 \left[(.915) \left(.045 - \frac{.0305}{2} \right) + .875 (.150) \right]$$

$$= 1.37 \times 10^6 (.1582)$$

$$= 217000 \text{ psi}$$

FOR L2 C6 718 - H.7-A, $F_{by} = 1.5 \times F_{ty} = 1.5 \times 150k$
 $= 225000 \text{ psi}$

$$\underline{M.S.} = \frac{225}{217} = 1.04$$

$$\underline{h = .3 \times .15 = .045'' \rightarrow .038 \text{ MIN}}$$

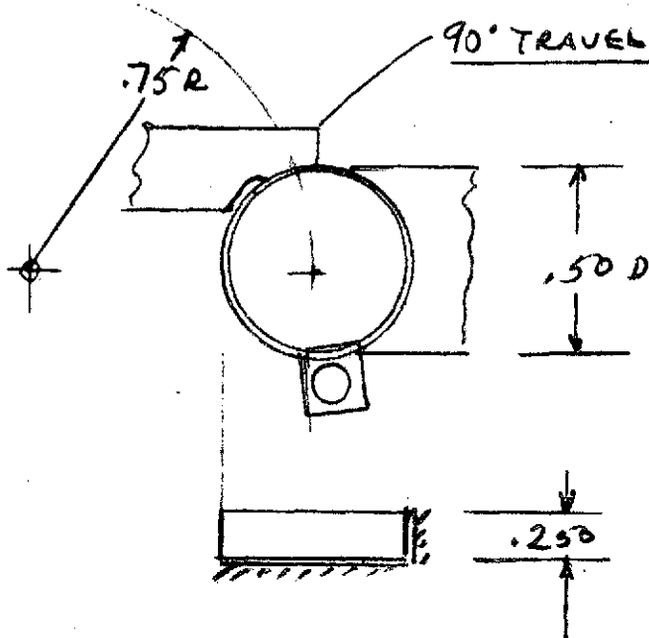
SHEAR ON LIP (ALUM 6061 T6)

$$A_s = \pi 3.28 (.09) = .930$$

$$\underline{S_s} = \frac{4280}{.930} = \underline{4600 \text{ psi}} \text{ SHEAR}$$

OME BALL VALVE

STOP SYSTEM - CLOSING VALVE



TORQUE MOTOR INERTIA .001 IN⁴ SEC²

SPEED BEFORE STOPPING = 100 RAD/SEC ON MOTOR

GEAR RATIO = 21.45 FOR CLOSING VALVE

TORSION SPRINGS ADD 2 x 12 IN-LBS TO STOP FORCE

KE OF MOTOR = $\frac{1}{2} I \omega^2 = .001 (100)^2 / 2 = 5.00 \text{ IN} \cdot \text{LBS}$

AT STOP, KE = 5.00 IN·LBS

SPRING FORCE 2 x 12 = 24.0 IN·LBS

OME BALL VALVE

STOP SYSTEM - CLOSING VALVE

KE FROM TORQUE MOTOR 500 IN# AT STOP

TORQUE FROM SPRINGS = 24.0 IN#

ASSUME ENERGY ABSORPTION AS:

$$\Delta \text{TORQUE} = \frac{I \omega^2}{2 \theta}$$

$$\theta = \frac{S}{R}$$

$$\Delta \text{TORQUE} = \Delta PR$$

$$\Delta PR = \frac{I \omega^2}{2 S/R}$$

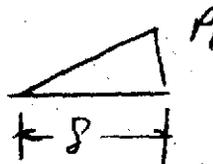
$$\Delta P = \frac{I \omega^2}{2 S}$$

$$\frac{I \omega^2}{2} = KE$$

$$\Delta PS = KE$$

BUT IF WE ASSUME:

$$\frac{\Delta PS}{2} = KE$$



TOTAL P_{MAX} = P₁ + 32.0 LBS

1/2 RING DEFLECTION - DIAMETRAL

$$S = 1.57 \frac{PR^3}{EI}$$

$$R = .625; R^3 = .245$$

$$I = \frac{.3125 (.125)^3}{12} = 50.8 \times 10^{-6}$$

$$S = \frac{1.57 (P) (.245)}{16 \times 50.8}$$

$$E = 16 \times 10^6 \text{ (7.)}$$

$$= .00473 P$$

FOR $t = .125$

OME BALL VALUESTOP SYSTEM

$$KE = \frac{1}{2} P \delta$$

$$5.00 = \frac{1}{2} (1.000473 P^2)$$

$$P^2 = \frac{2 \times 5.00}{1.000473} = 21100$$

$$P = 145.5 \times 32 = 177.5 \text{ \#}$$

MAX MOMENT ON RING = PR

$$M_{MAX} = 177.5 \times .625 = 111 \text{ in}^{\#}$$

$$f_b = \frac{6 \times 111}{.3125 (.125)^2} = 137000 \text{ psi MAX BENDING}$$

$$f_c = \frac{177.5}{.3125 \times .425} = 4550 \text{ psi}$$

$$\text{COMBINE } f_b + f_c = 141600 \text{ psi}$$

FROM MIL-HANDBOOK 5, FIG. 5.4.6.3.8(a)

$$N = 2 \times 10^4 \text{ cycles}$$

FOR $F_{TY} = 148 \text{ KSI}$ CAL-4U

$$R_b = \frac{137000}{112 \times 148000} = .76$$

$$R_c = \frac{4550}{148000} = .0307$$

COMBINE $R_b + R_c$ ACCORDING TO FIG 1d, LOCKHEED SM 53c

$$\text{SHORT TIME H.S.} = \frac{4.95}{3.8} - 1 = 1.30$$

OME BALL VALVE

STOP SYSTEM - GEAR TRAIN

SHAFT GEAR

1.8437 P.D.

GEAR RATIO (21.45:1)
(OVERALL)

32 DP

59 TEETH

STOP LOAD ON TOOTH

FROM STOP, $P = 177.5 \#$ (p 3)

MOMENT ABOUT GEAR $\Sigma = 177.5 \times .75 = 133 \text{ in} \#$

$$\text{LOAD TO TOOTH} = \frac{M_{\Sigma}}{PD/2} = \frac{133}{1.8437/2} = \underline{144.0 \#} = P$$

PITCH LINE VELOCITY ≈ 0 AT MAX LOAD.

CONTACT RATIO ON GEAR

$N_g = 59$ TEETH

$N_p = 19$ TEETH

FROM FIG 11, pg 950, MARKS - $R = 1.66$

MAX LOAD ON TOOTH CENTER F_{TC}

$$F_{TC} = P(1.4 - .4R) = 144(1.4 - .4(1.66)) = \underline{106 \#}$$

MAX LOAD ON TOOTH TIP F_{TT}

$$F_{TT} = P(R - 1).40 = 144(1.66 - 1)(.4) = \underline{38.0 \#}$$

DICKENS

7 NOV 1973

5 OF

OME BALL VALVESTOP SYSTEM - GEAR TRAINSHAFT GEAR

$$\begin{aligned} \text{TOOTH LOAD} &= \underline{106 \# \text{ AT CENTER}} = F_{TC} \\ &= \underline{38 \# \text{ AT TIP}} = F_{TT} \end{aligned}$$

BENDING STRESS (LOAD AT CENTER)

$$\sigma_b = \frac{W_p}{P y}$$

$$\text{LET } W_p = \frac{F_c}{1''}$$

$$\begin{aligned} P &= \pi \left(\frac{P.D.}{N} \right) = \pi \left(\frac{1.8437}{59} \right) \\ &= .0983 \end{aligned}$$

$$y = .225$$

$$\sigma_b = \frac{106}{.0983 \times .225} = 4800 \text{ psi FOR } 1'' \text{ WIDTH}$$

BENDING STRESS FOR LOAD AT TIP

$$y = .133$$

$$\sigma_b = \frac{38}{.0983 \times .133} = 2910 \text{ psi FOR } 1'' \text{ WIDTH}$$

FOR $F_p = 25000 \text{ psi}$ FOR 10^7 CYCLES - NOTCHED ACUM. ⁷⁰⁷⁵

(FULLY REVERSED)

FOR UNIDIRECTIONAL $F_{end} = 30000 \text{ psi}$

$$\underline{\text{MINIMUM WIDTH}} = \frac{4800 \times 1.5}{30000} \quad (1.5 \text{ SHOCK FACTOR})$$

$$= \underline{.25 \text{ IN}} \quad \text{FOR } 7075-76$$

(Pg 8 - MIN $t = .32$)

OME BALL VALVE

STOP SYSTEM - GEAR TRAIN

SHAFT GEAR

SHEAR STRESS ON TOOTH

$$\text{MAX LOAD} / \text{IN} = 106 \text{ #} / .25 = 424 \text{ #} = W$$

$$\text{STRESS} = W / \pi P.D. / N \times 2 \quad \left(\frac{\pi P.D.}{N} = .0983 \right)$$

$$\sigma_s = \frac{424}{.0983 / 2} = 8420 \text{ psi}$$

$$F_{\text{END}} = .60 \times 30000 = 18000 \text{ psi}$$

$$\frac{M.S. = 18000}{1.5 \times 8420} - 1 = .42$$

COMPRESSION STRESS ON TOOTH

$$S_c = .591 \sqrt{W E \frac{(d_1 + d_2)}{d_1 d_2}}$$

$$d = P.D \sin \theta$$

$$\theta = 20^\circ$$

$$\sin \theta = .342$$

$$d_1 = .342 (1.6432)$$

$$= .631$$

$$d_2 = .342 (.5937)$$

$$= .203$$

$$S_c = .591 \sqrt{424 (10^7) \frac{(.834)}{(.128)}}$$

$$S_c = 97500 \text{ psi MAX}$$

$$\text{MEAN STRESS} = 97500 / 2 = 48750 \text{ psi} = \text{ACT STRESS}$$

DICKENS

BALL VALVE

CHECKED BY

CLASSIFICATION

DATE

7 Nov 1973

PAGE

7

OF

OME BALL VALVESTOP SYSTEM - GEAR TRAINSHAFT GEARCOMPRESSION STRESS ON TOOTH

7075-76

FOR .25 WIDTH, $S_{C MAX} = 97500$ PSIMAX ALLOW $S_{C MAX} = 114000$ PSI - 7075-76

MIL-HANDBOOK-5A

$$\text{YIELD M.S.} = \frac{114000}{97500} - 1 = .17$$

FOR ENDURANCE AT 10^4 CYCLES - 7075-76

FEND = 1.8 FEND UN NOTCHED

$$= 1.8 \times 51000$$

$$= 91800 \text{ PSI}$$

ALT-STRESS = 48750 PSI

$$\text{10}^4 \text{ cycle END M.S.} = \frac{91800}{1.5 \times 48750} - 1 = .26$$

OME BALL VALVE

STOP SYSTEM - GEAR TRAIN

P07 PINION

7075-T6 MATL

.5937 P.D

32 DP

19 TEETH

TOOTH LOAD = 106 # AT CENTER (19 #)

FOR .250 WIDTH

$$W_p = 106 / .25 = 424 \# / \text{in} \quad (W_L = 106 / .32 = 332 \# / \text{in})$$

BENDING STRESS ON TOOTH

$$\sigma_b = W_p / p_y$$

$$p = \pi (P.D) / N = \pi (.5937) / 19 = .098$$

$$y = .170$$

$$\sigma_b = 424 / (.170 (.098))$$

$$\sigma_b = 25400 \text{ psi} \quad \text{MAKE } t = \frac{25400}{20000} \times .25 = .318 \text{ MIN}$$

$F_{end} = 30000 \text{ psi}$ MAX UNIDIRECTIONAL FOR 10^4 CYCLES
(MIL-HANDBOOK 5A)

FIG. 3.3.1 (J)

$$M.S. = \frac{30000}{1.5 \times 20000} - 1 = .00$$

SHEAR STRESS

$$\tau_s = W_p / \left(\frac{\pi P.D}{N \times 2} \right) \times \frac{.098}{2} = \frac{332}{.098/2} = 6760 \text{ psi}$$

$$F_{end} = .6 \times 30000 = 18000 \text{ psi @ } 10^4 \text{ cycles}$$

$$M.S. = \frac{18000}{1.5 \times 6760} - 1 = .77$$

DICKENS

BALL VALUES

8 NOV 1973

9

OME BALL VALUESTOP SYSTEM - GEAR TRAINGEAR + MOTOR PINION

$$\text{TORQUE TO GEAR} = 144.0 \times .5937/2 = 42.8 \text{ LBS} \cdot \text{IN}$$

(FROM TOTAL TOOTH LOAD ON SHAFT GEAR)

TOOTH LOAD TO GEAR + PINION

$$W_{TOT} = T/PD/2 = \frac{42.8 \times 2}{1.583} = 54.1 \# = P$$

GEAR 1.5833 P. D

48 DP

76 TEETHCONTACT RATIO ON GEAR $N_G = 76 \text{ TEETH}$ $N_P = 11 \text{ TEETH}$ FIG 11, P 9950, MARKS, $R = 1.18$ MAX LOAD ON TOOTH CENTER F_{TC}

$$F_{TC} = P(1.4 - .4R) = 54.1(1.4 - .4(1.18)) = 50.2 \#$$

MAX LOAD ON TOOTH TIP F_{TT}

$$F_{TT} = P(R - 1).40 = 54.1(1.18 - 1).40 = 3.9 \#$$

PREPARED BY
DICKENS

THE MARQUARDT COMPANY

REPORT
OME
BALL VALUES

CHECKED BY

CLASSIFICATION

DATE
8 Nov 1973

PAGE
10 OF

OME BALL VALUE

STOP SYSTEM - GEAR TRAIN

GEAR & MOTOR PINION

7075-T6 MATL

GEAR TOOTH LOADS = 50.2 # AT CENTER
= 3.9 # AT TIP

BENDING STRESS (LOAD AT CENTER) (WIDTH = 1" ASSUME

$W_p = 50.2 \text{ #/N}$
 $\sigma_b = W_p / P_y$
 $P = \pi \frac{(PD)}{N} = \frac{\pi 1.5833}{76}$
 $= .0656$
 $y = .234$

$\sigma_b = 50.2 / (.234 \times .0656) = 3265 \text{ psi FOR } 1"$

FOR FEND = 30000 psi MAX STRESS FOR 10⁷ CYCLES

$WIDTH = \frac{3265 \cdot 1.5}{30000} \times 1 = .163 \text{ in (700 NARROW)}$

USE 1.25 WIDTH - BENDING NOT CRITICAL *

COMPRESSION STRESS ON TOOTH

$S_c = .591 / WE \left(\frac{d_1 + d_2}{d_1 d_2} \right)$
 $= .591 \left[200.8 \left(\frac{.6204}{.1045} \right) 10^7 \right]^{1/2}$
= 10/000 psi MAX

$d = PD \sin \theta$
 $\sin \theta = \sin 20^\circ = .342$
 $d_1 = 1.5833 \times .342 = .542$
 $d_2 = .2291 \times .342 = .0784$
 $WE = 50.2 / .25 = 200.8$
 $E = 10^7$

MAKE $t = .318 \text{ MIN (1211)}$

$S_c = \left(\frac{.25}{.318} \right)^2 \times 10/000 = 89600 \text{ psi COMPRESSION MAX}$

OME BALL VALUE

STOP SYSTEM - GEAR TRAIN

MOTOR PINION

AI. 7075-76

.2291 PD

48 DP

11 TEETH

FOR .25 WIDTH

$$W_c \text{ MAX} = \frac{5012}{.25} = 20048 \text{ } \frac{\text{lb}}{\text{in}}$$

BEADING STRESS ON TOOTH

$$\sigma_b = \frac{W_c}{p_y}$$

$$p = \frac{\pi PD}{D}$$

$$= \frac{\pi (.2291)}{11} = .0654$$

$$y \approx .122$$

$$\sigma_b = \frac{20048}{.122 \times .0654} = 25200 \text{ psi}$$

HAKE t = .318 MIN

$$W_c = \frac{5012}{.318} = 15761 \text{ } \frac{\text{lb}}{\text{in}}$$

$$\sigma_b = \frac{.25}{.318} \times 25200 = 20000 \text{ psi}$$

$F_{end} = 30000 \text{ PSI MAX UNIDIRECTIONAL FOR } 10^4 \text{ CYC.}$

$$\underline{\underline{M.S. = \frac{30000}{1.5 \times 20000} - 1 = .00}}$$

SHEAR STRESS = $\frac{W_c}{\frac{\pi PD}{D+2}}$

$$= \frac{15761 \times 2}{.0654} = 4840 \text{ psi}$$

$$F_{end} = .6 \times 30000 = 18000 \text{ psi}$$

$$\underline{\underline{M.S. = \frac{18000}{1.5 \times 4840} - 1 = 1.48}}$$

PREPARED BY
DICKENS

THE MARQUARDT COMPANY

REPORT
OME
BALL VALUES

CHECKED BY

CLASSIFICATION

DATE

9 Nov 1973

PAGE

12 OF

OME BALL VALUES

STOP SYSTEM - GEAR TRAIN

MOTOR PINION

7075-76

MAX COMPRESSION STRESS = 89600 psi

FOR .32 WIDTH

YIELD BEARING STRESS = 114000 psi (MIL-HANDBOOKS)

$$\text{YIELD M.S.} = \frac{114000}{89600} - 1 = .26$$

ENDURANCE (COMPRESSION)

FEND @ 10⁷ CYCLES = 1.8 × 51000 = 91800 psi

UNNOTCHED

ALTERNATING STRESS = 89600 / 2 = 44800 psi

$$\text{10⁷ CYCLE LIFE M.S.} = \frac{91800}{1.5444800} - 1 = .36$$

OME BALL VALVE

SHAFT, X29649

GAC-4VTI ALLOY

MAX STOP LOAD = 177.5 LB (#3)

$F_{T0} = 150K$
 $F_{TY} = 120K$

$T_g = P \times R = 177.5 \times .75 = 133 \text{ IN} \cdot \#$

LOAD TO GEAR PIN, R = .625

REACTION = $133 / .625 = 213 \#$

PIN IS .183 IN. MIN. DIA

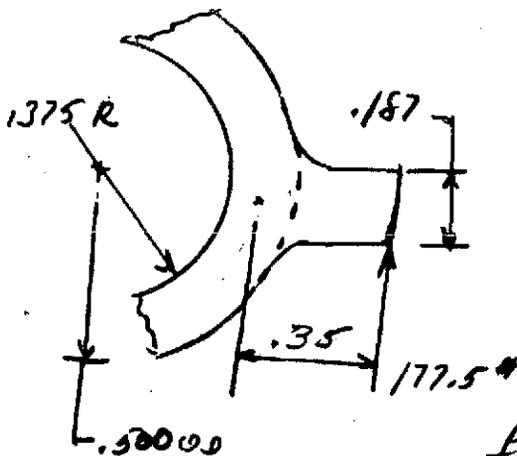
$A_s = \frac{\pi}{4} (.183)^2 = .0263 \text{ IN}^2$

$f_s = 213 / .0263 = 8100 \text{ PSI SHEAR}$

$F_{SY} = .6 \times 120,000 = 72,000 \text{ PSI}$

M.S. = LARGE

BENDING ON SHAFT WALL



ASSUME MOMENT REACTED BY RING .125 DEEP

$M_{O/SIDE} = 177.5 \times .35 / 2 = 31.1 \text{ IN} \cdot \#$

LET $I_y = \frac{.125 (.125)^3}{6} = .000325$
(CONSERVATIVE)

$f_b = 31.1 / .000325 = 94300 \text{ PSI}$

$F_{TY} = 120,000 \text{ PSI}$

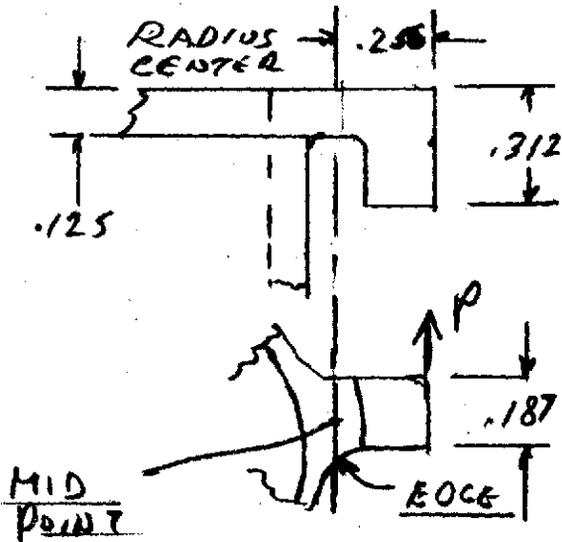
$F_{BY} = 1.22 \times 120,000 = 146,000 \text{ PSI}$

M.S. = $\frac{146,000}{94,300} = 1.55$

OME BALL VALVE

SHAFT X 29649

BENDING & TORSION ON STOP LOG



$$M = 177.5 \times .25 = 44.4 \text{ in} \cdot \text{lb}$$

$$T_2 = 177.5 \left(\frac{.312}{2} - \frac{.125}{2} \right)$$

$$T_2 = 18.6 \text{ in} \cdot \text{lb}$$

$$f_b = \frac{6M}{b t^2} = \frac{6(44.4)}{.125(.177)^2} = \underline{68000 \text{ psi AT EDGE}}$$

$$\tau = \frac{3T_2}{b t^2} = \frac{3(18.6)}{.125(.177)^2} = \underline{12700 \text{ psi AT MID POINT}}$$

FATIGUE

FOR MAX STRESS = 94300 psi UNI-DIRECTIONAL
PS 13

ALLOWABLE N = 7 10⁷ cycles

REF MIL-HANDBOOK 5A

FIG 5-4.6.2.8(a) ANN. 6AL-4V

H.S. - LARGE

THIS PAGE LEFT INTENTIONALLY BLANK

THE

Marquardt
COMPANY

VAN NUYS, CALIFORNIA

APPENDIX C

**PARTS LIST
PROTOTYPE BALL VALVE**

THIS PAGE LEFT INTENTIONALLY BLANK

PARTS LIST													MODEL NO.	PAGE 1 of 4	PART NO. X 29630				
MAC 81388													CONTRACT NO.	NEXT ASSY.	TITLE BALL VALVE, PROTOTYPE				
LINE	PARTS BREAKDOWN													CHG. LTR.	PART NAME OR DESCRIPTION	REQ. N/A	REMARKS		
	1	2	3	4	5	6	7	8	9	10	11	12	13					14	15
1	X29630																BALL VALVE ASSY	1	
2																			
3	X29631-1																HOUSING & SUPPORT ASSY	1	
4	X29631-3																HOUSING	1	
5	X29639-1																HOUSING, WELDED	1	
6	X29639-5																BODY	1	
7	X29639-7																BOSS	1	
8	X29639-11																PAD	1	
9	X29631-7																SUPPORT	1	
10	X29632-3																SUPPORT	1	
11	X29632-5																BUSHING	2	
12	3585-04C10168																INSERT, HELI-COIL	2	
13	3591-3CN190																INSERT, HELI-COIL	12	
14	X29631-																PIN	2	
15	X29631-501																HOUSING & SUPPORT ASSY	1	
16	X29631-5																HOUSING	1	
17	X29639-501																HOUSING, WELDED	1	
18	X29639-5																BODY	1	
19	X29639-9																BOSS	1	
20	X29639-13																PAD	1	
21	X29631-9																SUPPORT	1	
22	X29632-3																SUPPORT	1	
23	X29632-5																BUSHING	2	
24	3591-3CN190																INSERT, HELI-COIL	12	
25	3585-04C10168																INSERT, HELI-COIL	2	
26	X29633																BALL, VALVE	2	
27	X29633-3																BALL	1	
28	X29633-5																BEARING	2	
29																			
30	X29634																FITTING, SHAFT SEAL	2	
31	X29634-3																PAD	1	
32	X29634-5																TUBE	1	
33																			
34	X29635																SPACER, BALL VALVE	4	
35																			
36	X29636																FILTER, BALL VALVE	2	
37	X29636-5																CLAMP	1	
38	X29636-5																SCREWS	1	
39	X29636-7																RING	1	
40																			
1	2	3	4	5	6	7	8	9	10	11	12	13	14	15	16	CHG. LTR.	DESCRIPTION	REQ. N/A	REVISION DATE



PARTS LIST																MODEL NO.	PAGE 2 of 4	PART NO. X29630		
MAC 0130																CONTRACT NO.	NEXT ASSY.	TITLE BALL VALVE, PROTOTYPE		
L I N E	PARTS BREAKDOWN																CHG. LTR.	PART NAME OR DESCRIPTION	REQ. N/A	REMARKS
	1	2	3	4	5	6	7	8	9	10	11	12	13	14	15	16				
1	X29630																	CONT'D		
2	X29637																	SPRING, RETAINING	2	
3																				
4	X29638																	RETAINER, SPRING	2	
5																				
6	X29640																	SOLENOID ASSY	2	
7	X29640-3																	RETAINER	1	
8	X29640-5																	SEAL	1	
9	X29640-7																	SOLENOID	1	
10	X29642																	SOLENOID, BALL VALVE	1	
11		X29642-3																BODY	1	
12		X29641																BODY, SOLENOID	1	
13		X29642-5																FLEXURE	1	
14		X29643																FLEXURE ASSY.	1	
15		X29643-2																TUBE	1	
16		X29643-5																DIAPHRAM	2	
17		X29643-7																RING	1	
18	X29640-9																	ARMATURE	1	
19	X29644																	ARMATURE	1	
20	X29645																	ARMATURE, BRAZED	1	
21		X29645-3																ARMATURE	1	
22		X29645-5																SEAT	1	
23																				
24	X29646																	CLOSURE	1	
25		X29646-3																RING	1	
26		X29646-5																RING	1	
27		X29646-7																DISC	1	
28																				
29	X29647																	COIL ASSY	1	
30																				
31																				
32	X29648																	CHANNEL	1	
33																				
34	X29649-1																	SHAFT	1	
35																				
36	X29649-501																	SHAFT	1	
37																				
38																				
39																				
40																				
1	2	3	4	5	6	7	8	9	10	11	12	13	14	15	16	CHG. LTR.	DESCRIPTION	REQ. N/A	REVISION DATE:	

PARTS LIST																MODEL NO.	PAGE 3 OF 4	PART NO. X29630		
MAC 8122																CONTRACT NO.	NEXT ASSY.	TITLE BALL VALVE, PROTOTYPE		
LINE	PARTS BREAKDOWN																CHG. LTR.	PART NAME OR DESCRIPTION	REQ. N/A	REMARKS
	1	2	3	4	5	6	7	8	9	10	11	12	13	14	15	16				
1	X29630																	CONT'D		
2																				
3	X29650																	HOUSING ASSY	1	
4	X29650-3																	HOUSING	1	
5	MS9390-170																	DOWEL	2	
6	3585-3CNDV100																	INSERT, HELI-COIL	6	
7	3585-D2CND124																	INSERT, HELI-COIL	4	
8	3585-D2CND172																	INSERT, HELI-COIL	16	
9																				
10	X29651																	COVER ASSY	1	
11	X29651-3																	COVER	1	
12	3585-3CNDV100																	INSERT, HELI-COIL	2	
13																				
14	X29652																	COVER, MOTOR		
15																				
16	X29653-1																	SHAFT GEAR	1	
17																				
18	X29653-501																	SHAFT GEAR	1	
19																				
20	X29654-1																	GEAR, POTENTIOMETER	1	
21	3585-D2CND129																	INSERT, HELI-COIL	1	
22																				
23	X29654-501																	GEAR, POTENTIOMETER	1	
24	3585-D2CND129																	INSERT, HELI-COIL	1	
25																				
26	X29655																	PINION, MOTOR	1	
27																				
28	X29656																	COVER, POTENTIOMETER	1	
29																				
30	X29657																	SPRING, TORSION	2	
31																				
32	X29658																	SPRING, STOP	2	
33																				
34	X29659																	PLATE, POTENTIOMETER	1	
35																				
36																				
37																				
38																				
39																				
40																				
	1	2	3	4	5	6	7	8	9	10	11	12	13	14	15	16	CHG. LTR.	DESCRIPTION	REQ. N/A	REVISION DATE

PARTS LIST															MODEL NO.	PAGE 4 of 4	PART NO. X 29630			
															CONTRACT NO.	NEXT ASSY.	TITLE BALL VALVE, PROTOTYPE			
LINE #	PARTS BREAKDOWN															CHG. LTR.	PART NAME OR DESCRIPTION	REQ. N/A	REMARKS	
	1	2	3	4	5	6	7	8	9	10	11	12	13	14	15					16
1	X29660																	MOTOR	1	
2																				
3	M521043-3																	NUT	2	
4																				
5	M516996-16																	SCREW	4	
6	M516996-13																	SCREW	2	
7	M516996-11																	SCREW	18	
8	M516996-10																	SCREW	2	
9	M516996-9																	SCREW	6	
10																				
11	M516995-9																	SCREW	8	
12	M516995-2																	SCREW	16	
13																				
14	M516562-190																	PIW	4	
15																				
16	M52112	12																CONNECTOR	1	
17																				
18	AR10105-233A1Q																	OMNISEAL	4	
19	AR10105-209A1Q																	OMNISEAL	2	
20	AR10105-207A1Q																	OMNISEAL	2	
21	AR10105-205A1Q																	OMNISEAL	2	
22	AR10105-203A1Q																	OMNISEAL	2	
23	AR10105-106A1Q																	OMNISEAL	2	
24																				
25	M5939C-170																	DRWEL	2	
26																				
27																				
28																				
29																				
30																				
31																				
32																				
33																				
34																				
35																				
36																				
37																				
38																				
39																				
40																				

APPENDIX D
TEST PLAN MTP 0200

C-3

THIS PAGE LEFT INTENTIONALLY BLANK

OME BALL VALVE EVALUATION TESTS

1.0 OBJECTIVE

The objective of this test plan is to describe tests and procedures to evaluate performance of the OME ball valve. The required performance characteristics include response, pressure drop, leakage and wear characteristics.

2.0 DESCRIPTION OF TEST HARDWARE

The test hardware consists of a bipropellant electrically actuated ball valve defined by drawing X29630. In addition to the brushless DC torque motor used for valve opening actuation, the valve also contains two solenoids actuated to lift the seal off of the ball prior to ball rotation. The electronic driver P/N X29663 for the brushless torque motor performs the necessary current switching to the motor coils by

1. Sensing armature position with two "Hall Effect" elements mounted on the stator.
2. Performing the necessary logic and signal conditioning and
3. Driving the proper phase of the motor windings with voltage square wave.

3.0 INSTRUMENTATION

<u>Parameter</u>	<u>Range</u>	<u>Accuracy</u>
Temperature	0-250°F	±5°F
Valve Position (2)	0-10V	±.2V
Internal Leakage		±1 scc/HR
Inlet Pressure	0-300 psig	±3 psi
Pressure Differential	0-1 psid	±.05 psi
Flow Meter	0-30 gpm	±.3 gpm
Torque	0-4 in #	±.2m #
RPM	0-1000 rpm	±10 rpm

<u>Parameter</u>	<u>Range</u>	<u>Accuracy</u>
Solenoids (2)	0-36	±.2 V
Voltage Meter	0-36	±.2 V
Solenoids (2)	0-1 amp	±.01 amps
Current Meter	0-3 amps	±.03 amps
No. Actuations	0-10,000	±1

NOTE: A strip chart recorder will be used to record valve positions and torque motor and solenoid currents and voltages to determine response characteristics.

4.0 TEST PROCEDURE

The general test procedure is to complete electronic component tests to check out driver circuit and verify torque motor and solenoid performance and then perform assembled valve tests to evaluate pressure drop, leakage, response and wear characteristics using the test setup per Figure 1.

4.1 MOTOR SPEED

Apply known voltage to torque motor and measure current and motor rpm under no load conditions. Perform test from 0 to 30 volts in increments of 5 volts.

4.2 MOTOR TORQUE

Apply known voltage to torque motor and measure current and motor output torque in stalled condition. Perform test from 0-35 volts in increments of 5 volts. Determine stall torque ripple by slowly turning motor and measuring

torque over 1/4 revolution.

4.3 CURRENT LIMITING AND DYNAMIC BRAKING

Set current limit to 1.6 amps using dummy load to prevent demagnetizing of permanent magnet armature. Adjust timing on dynamic braking to approximately TBD sec. Final adjustment will be made during response tests.

4.4 PROOF PRESSURE

With valve in closed position slowly pressurize upstream pressure to 375 psig and hold for one minute. With valve in open position and facility valve downstream closed slowly pressurize to 375 psig and hold for one minute. There should be no external leakage or evidence of permanent distortion or damage.

4.5 PRESSURE DROP

Open valve to wide open position and increase water flow rate through valve while recording water flow rate and pressure differential across valve. Perform test 0-30 gpm in increments of 10 gpm. Because of low level pressure drop, care must be exercised to bleed all air from gage lines and also prevent differential head between the two gage lines.

4.6 INTERNAL LEAKAGE

Purge dry with helium and close valve. Increase upstream helium pressure to 25 psig. Measure internal leakage by displaced liquid technique for period of six minutes. Repeat above procedure with upstream pressure at 250 psig.

4.7 VALVE RESPONSE

4.7.1 Opening Response versus Voltage

The total opening time (time from application of electrical signal to time valve impacts the stop) shall be determined by recording command voltage current, and valve positions. The response test shall be run at 15, 20, 25, 30, 35 Volts and the pressure shall be 250 psig.

TEST PLAN

MTP 0220

PAGE

4

OF 6

4.7.2 Opening Response versus Pressure

The total opening time shall be determined as a function of upstream water pressure. The applied voltage shall be 30 volts and runs at upstream pressure at 0, 200, 225, 250 psig shall be made.

4.7.3 Closing Response

The closing response (time from removal of electrical signal to time of impact on closing stop) shall be determined at 0, and 250 psig. Valve position signals and strip chart recorder shall be used to compute impact velocities.

4.7.4 Lead-Lag Characteristics

Both fuel and oxidizer valve positions shall be documented on a strip chart recorder during opening and closing response for evaluation of lead-lag characteristics.

4.8 THERMAL SOAK

Lay complete dry valve assembly in still air with electronic driver providing holding power at two conditions (20 and 30 volts to the valve). Measure and record temperatures at selected locations on the valve at 5 minute increments until steady state temp exists. Use contact tyro-meter for external temp measurement. Measure internal stator iron temp with thermocouple attached to stator.

TEST PLAN

MTP 0220

PAGE

5

OF 6

4.9 LIFE CYCLING

Total of 10,000 cycles (4000 wet and 6000 dry) shall be completed in four groups of 2500 cycles with 1000 wet cycles and 1500 dry cycles. The cycling shall be completed at a frequency of approximately 1000 cycles per hour, being run as individual groups. The wet cycles are run with water at 250 psi with a downstream throttling valve set to limit flow to about 1 gpm. After wet cycling (1000 cycles) the valve shall be dried out by flowing nitrogen through it for TBD minutes to thoroughly dry internal valve parts. The dry cycling will be completed (1500 cycles) with upstream nitrogen pressure at 5 psig.

Following each group of 2500 cycles the valve shall be tested to determine its performance degradation. A leakage check as described in Section 4.6 shall be conducted. An opening response test at 28V and 250 psia and a closing response at zero volts and zero pressure shall be conducted. If a severe change from prior tests in either leakage or response is evident during these tests, hold unit for project engineer disposition.

4.10 TEAR DOWN INSPECTION

At the conclusion of the life cycle testing, the valve shall be disassembled and visually inspected for wear. If excessive wear is noted, the worn components shall be dimensionally inspected to determine extent of the wear. Photographs of critical component parts shall be taken for comparison with pictures taken prior to initiation of tests.

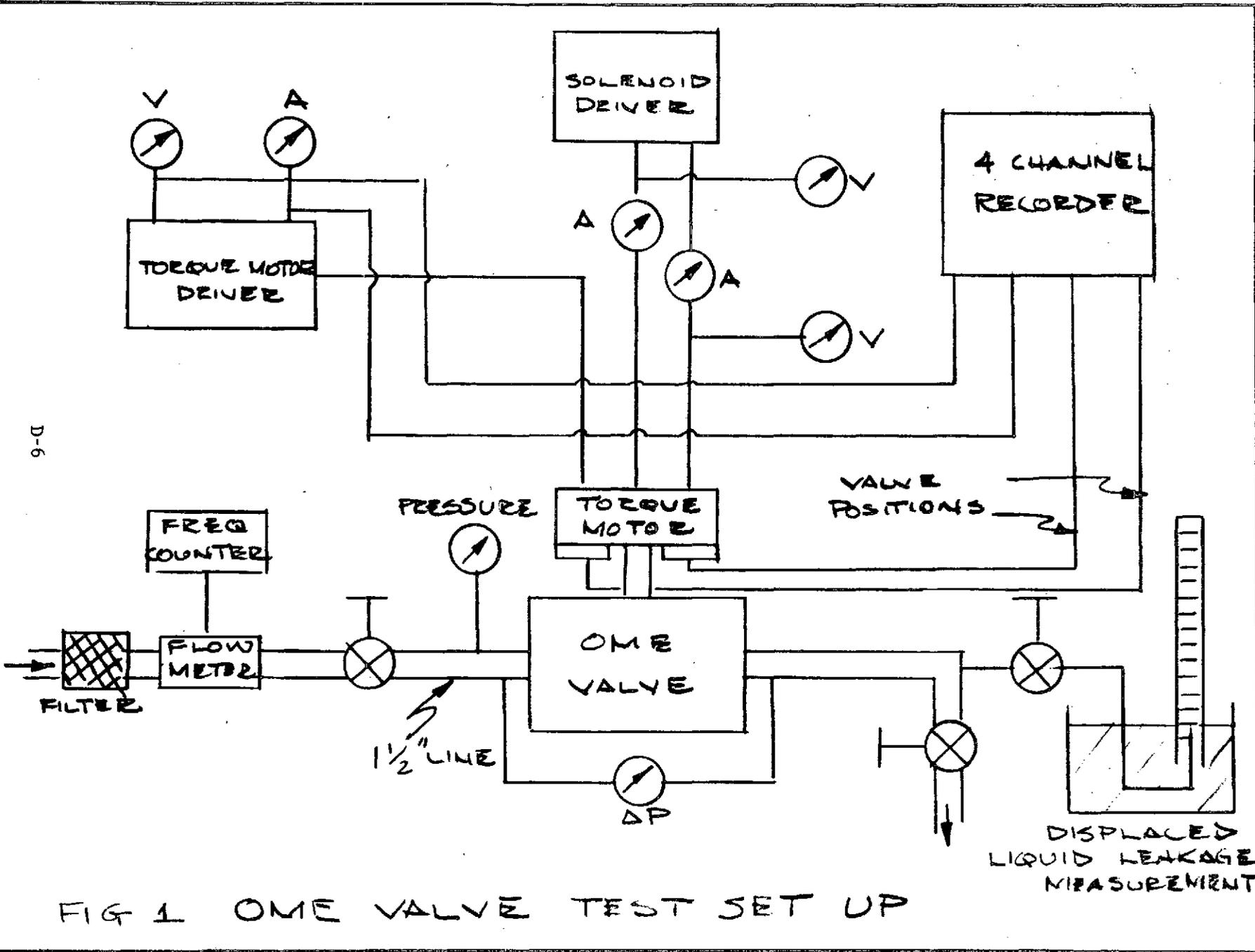


FIG 1 OME VALVE TEST SET UP

D-6